Carderock Division
Naval Surface Warfare Center
Bethesda, MD 20817-5700

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Total Ship Systems Engineering Directorate
Technology Projection Report

HIGH-SPEED SEALIFT TECHNOLOGY DEVELOPMENT PLAN

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High-Speed Sealift Technology Development Plan
Contents

High-Speed Sealift Technology Development Plan
Contents (continued)

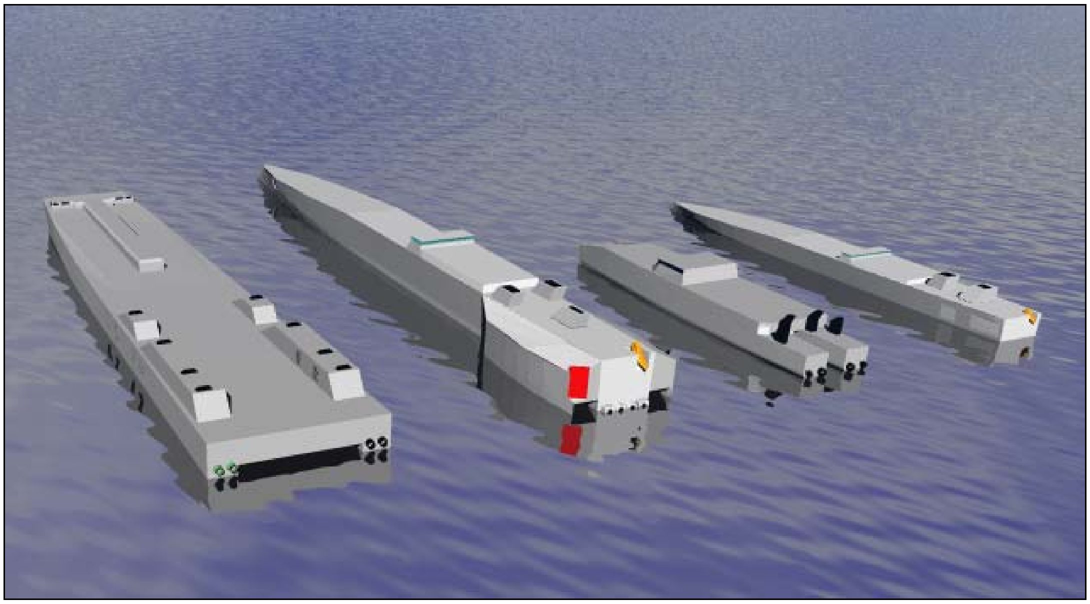
High-Speed Sealift Technology Development Plan
Contents (continued)

High-Speed Sealift Technology Development Plan
List of Figures

High-Speed Sealift Technology Development Plan
List of Figures (continued)

List of Tables

EXECUTIVE SUMMARY

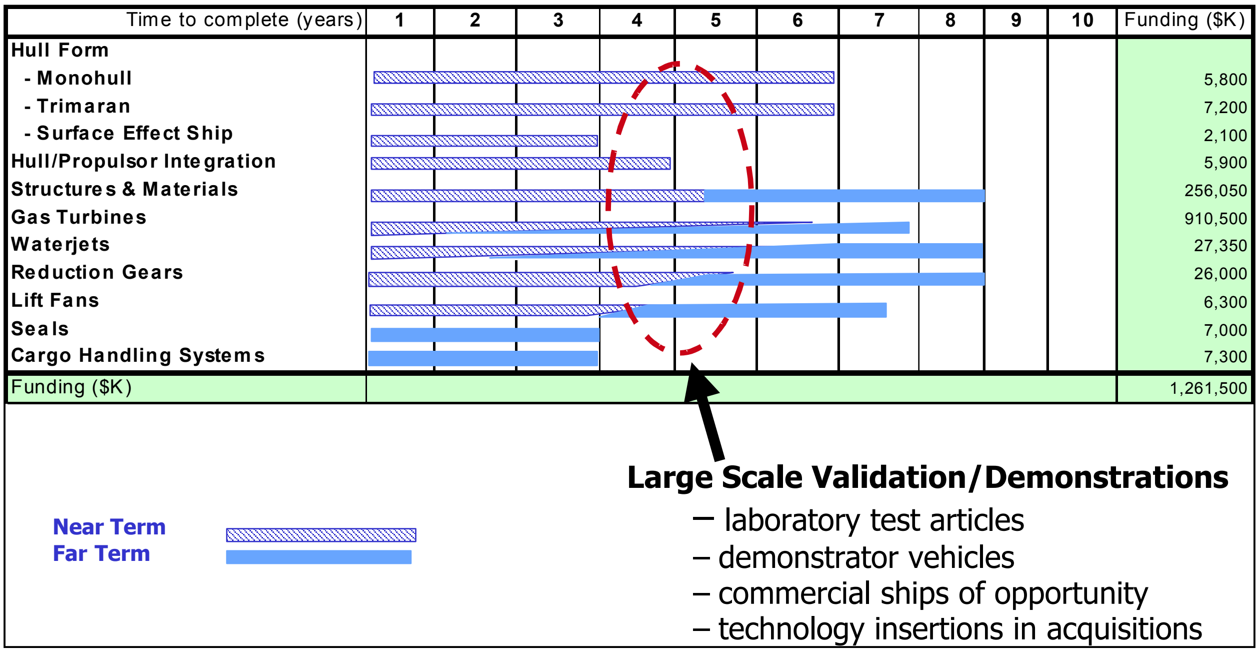


This document summarizes work performed by the High-Speed Sealift Innovation Cell project conducted at the Naval Surface Warfare Center, Carderock Division from May 2000 through August 2001. The purpose of the project was to define the technology investments required to enable development of the high-speed commercial and military ships needed to provide realistic future mission capabilities.

This project is part of a process initiated within the U.S. Department of Defense and industry to help define the next generation of sealift ships. A High-Speed Sealift Technology Workshop, sponsored jointly by U.S. Transportation Command in partnership with the Center for the Commercial Deployment of Transportation Technologies, U.S. Maritime Administration, U.S. Army/DCSLOG, and U.S. Navy/PEOCLA, was held at the Naval Surface Warfare Center, Carderock Division in October 1997 to examine the possibilities offered by technology to enhance the transport performance of high-speed commercial and military sealift ships. Technology projections were made in six key areas; namely, Ship/System Concepts, Hullforms and Propulsors, Propulsion Plant, Cargo Onload/Offload and Stowage, Materials and Ship Structures, and Shipbuilding and Manufacturing. The Workshop, combined with subsequent analysis, predicted levels of sealift capability associated with different technologies. Economic considerations were not introduced at this stage since the initial focus was on determination of technological feasibility without regard to cost of development or commercial viability.

Following the Workshop, a High-Speed Sealift Executive Steering Committee (HSSESC) was formed to coordinate U.S. Army, U.S. Transportation Command, and U.S. Navy HSS efforts. The High-Speed Sealift Innovation Cell project was chartered by the HSSESC to take this technology guidance and convert it into concept ships to examine the whole-ship implications of the technology. The main aim of the Innovation Cell was to derive a Technology Development Plan (TDP) on the basis of demonstrable need and platform performance pay-off. Technologies were classed as near-term (available in 5 years) and far-term (available in 10 years).

Action officers for the HSSESC provided nine hypothetical military and commercial missions. Monohull, catamaran, trimaran and Surface Effect Ship (SES) designs were produced to a uniform standard for each of the missions. Technology projections from the HSS Technology Workshop for structures and materials, gas turbines, reduction gears, and waterjets were combined with additional technical information to produce a common basis for these technologies in the designs.



The capabilities needed from each of the technologies to produce these designs were compared with the technical state-of-the-art for those technologies to define the necessary near-term and far-term technology enhancements. Estimates of the time to develop and rough order of magnitude development costs were made for each of the technologies based on a variety of factors including experience with development of similar technologies, engineering estimates, vendor data, and cost models. The goal of this plan is to bring the individual technologies to a level of maturity sufficient to lower risk to levels appropriate to ship design and construction.

This development plan is comprehensive, with no allowance for market-driven technology development that may occur through commercial initiatives. Some technology development in critical areas is expected to meet anticipated commercial needs for aerospace, industrial, and commercial marine projects. While such commercial technology development efforts will potentially reduce the need for Government investment, elimination of this investment is not expected since there is some risk that the commercial efforts will either not come to fruition or the commercially-derived capabilities will fall short of the capabilities needed to meet the more demanding military missions. Consequently, the potential existence of these commercial efforts is identified, while the cost reductions that might result have not been shown.

The plans contain some necessary redundancies since the specific need for some of the technologies depends on other technology choices. The choice of hullform technology has a particularly large impact on requirements for other technologies. For example, development of far-term SES hulls requires development of SES-specific lift fan and seal technologies. Alternately, monohull and trimaran hulls require development of locked-train, double-reduction (LTDR) gear technology, SES hulls require epicyclic reduction gear technology development, and catamarans require a mix of epicyclic and LTDR technologies. Since choices such as these cannot be made with certainty prior to commitment to specific long-term objectives, the redundancies have been identified and retained at the individual technology level. However, it is unlikely that the full matrix of technologies will be developed.

1.0 INTRODUCTION

A process has been initiated within the U.S. Department of Defense and industry to help define the next generation of sealift ships. A High-Speed Sealift Technology Workshop[1], sponsored jointly by U.S. Transportation Command in partnership with the Center for the Commercial Deployment of Transportation Technologies, U.S. Maritime Administration, U.S. Army/DCSLOG, and U.S. Navy/PEOCLA, was held at the Naval Surface Warfare Center, Carderock Division in October 1997. This Workshop examined the possibilities offered by technology to enhance the transport performance of high-speed (40-100 knots) commercial and military sealift ships, in advance of detailed design studies, in order to help define realistic future mission capabilities and to focus the subsequent design and cost studies necessary to enable technology investment decisions.

The Workshop solicited expert opinion to address technology projections in six key areas; namely, Ship/System Concepts, Hullforms and Propulsors, Propulsion Plant, Cargo Onload/Offload and Stowage, Materials and Ship Structures, and Shipbuilding and Manufacturing. Economic considerations were not introduced at this stage since the initial focus was on determination of technological feasibility without regard to cost of development or commercial viability. The Workshop, combined with subsequent analysis, developed predictions of expected levels of sealift capability associated with different technologies. Mission parameters speed, range, and payload were related to ship design characteristics displacement, installed power, cargo weight, and fuel weight. These were presented at Naval Surface Warfare Center, Carderock Division in March 1998 and subsequently published[2].

Figure 1-1 represents the maximum mission performance associated with the technology projections made at the Workshop. It shows that significant sealift capabilities are scientifically possible using such technology projections in the near-term and the far-term, where the near-term relates to technology that will be available in 5 years and the far-term, 10 years. Full realization of the sealift capabilities shown in Figure 1-1 requires engineering development, particularly in packaging propulsion technology, advanced hullforms, and advanced materials and structures.

A wide range of options was identified by the workshop. While the impact on transport capability of some of the more significant of these technologies has been defined, the required mix of technologies and their expected cost depend on specific mission requirements such as speed, range and payload. Detailed design studies are needed to make the necessary technology investment selections. The process shown in Figure 1-2 was developed to guide determination of the needed technology investments.

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[1] references are at the end of each section

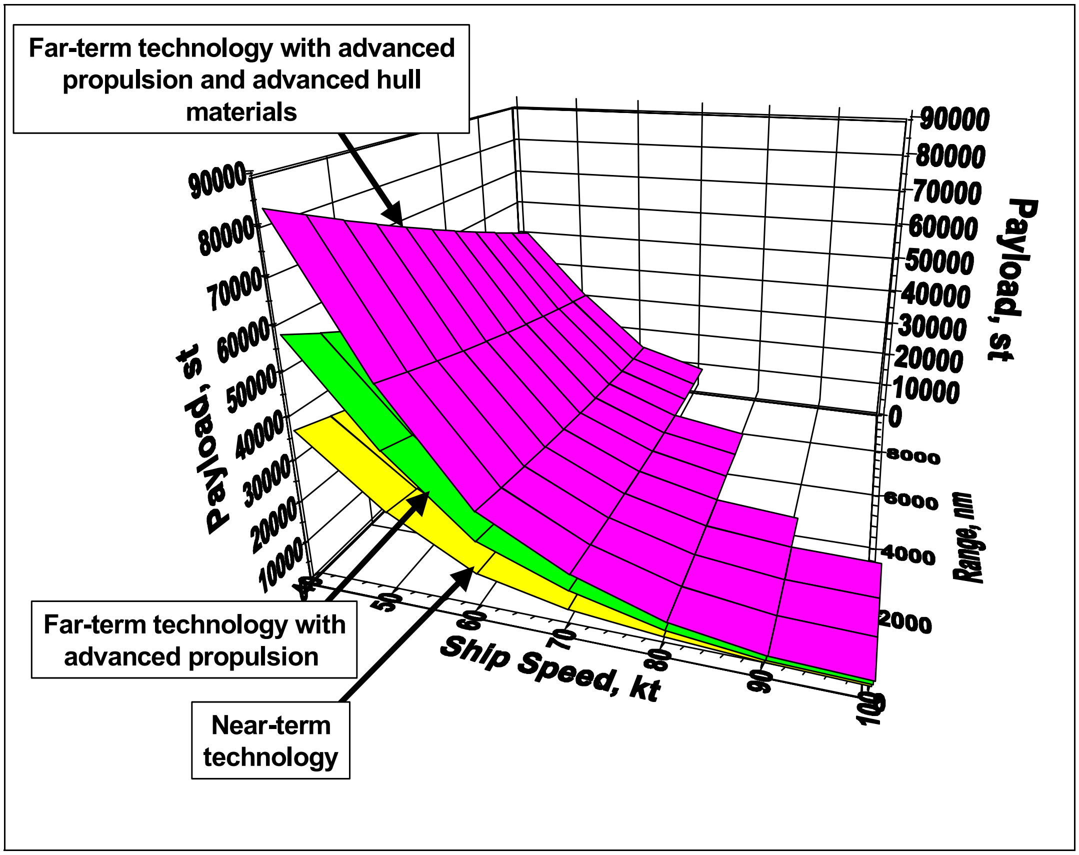


Figure 1-1: Predicted Impact of Technology on Ship Performance

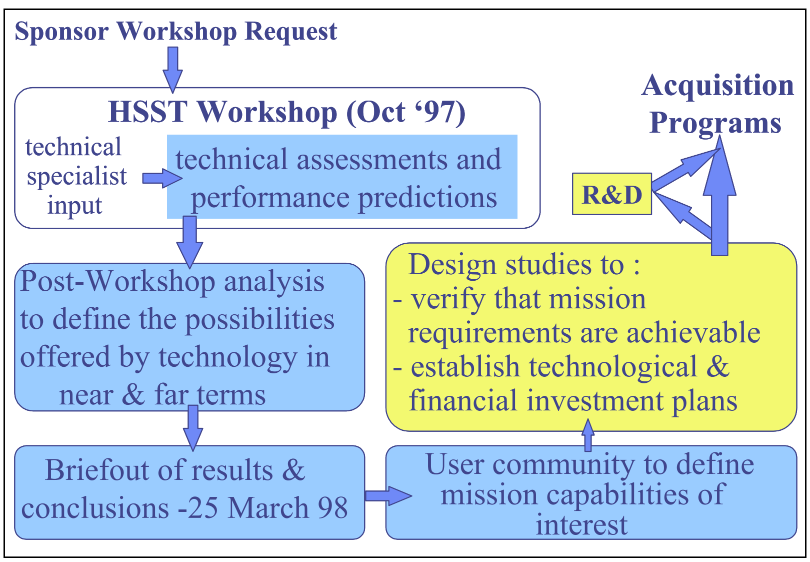


Figure 1-2: Technology Development Process

Following the workshop, an Executive Steering Committee (ESC) was formed to co-ordinate U.S. Army, U.S. Transportation Command, and U.S. Navy HSS efforts. The High-Speed Sealift Innovation Cell project was chartered by the ESC to take this technology guidance and convert it into concept ships to examine the whole-ship implications of the technology. The main aim of the Innovation Cell was to derive a Technology Development Plan (TDP) on the basis of demonstrable need and platform performance pay-off. Technologies were classed as near-term (available in 5 years) and far-term (available in 10 years). There were also potential spin offs from the designs developed including technology pointers as to key factors to particular missions, realistic ship concepts for operational analysis and planning, and creation of a basis for discussions with other organizations and exploration of their interest in research involvement and developing technologies.

Action officers for the ESC provided nine hypothetical military and commercial missions. Sensitivity studies around range, speed, and payload generated another additional six design points. In addition to mission parameters of speed, range, and payload, each mission description included a technology characterization.

Included in the mission set are both short range coastal commercial/intra-theater military missions as well as long range trans-ocean commercial/inter-theater military missions. Speeds for the shorter range missions were relatively low (40-50 knots). Higher speeds (55-70 knots) were specified for the longer ranges associated with the inter-theater/trans-ocean missions. Payloads varied from a few hundred tonnes for the least demanding intra-theater mission to 12,000 tonnes for the most demanding inter-theater missions. The missions are summarized in Table 1-1.

Table 1-1: Mission Summary

	Shuttle Ship 1a	Shuttle Ship 1b	Intra-Theater Support Ship 2a	Intra-Theater Support Ship 2b	Coastal Commercial Ship 3	Trans-Ocean Commercial Ship 4a	Trans-Ocean Commercial Ship 4b	Inter-Theater Ship 5
Average Speed (knots)	40	45	40	40	50	50	60	40
Full Performance Wave Height (m)	2.4	2.4	2.4	2.4	2.4	4	4	4
Range (nm)	1,250	1,250	800	1,200	1,500	4,000	4,000	5,000
Payload (mt)	1,497	1,497	454	454	1,500	7,500	7,500	5,445
Ramp Requirements	y	y	y	y	n	n	n	y
Total Crew	20	20	20	20	20	30	30	30
Structural Technology	current	current	current	current	current	far	far	near
Waterjet Technology	current	current	current	current	current	far	far	near
Prime Mover Technology	current	current	current	current	current	far	far	near

	Vision Ship 70 knots 6a	Vision Ship 60 knots 6b	Vision Ship 55 knots 6c	Vision Ship 5,000 st 7a	Vision Ship 7,500 st 7b	Intra-theater Ship 8	Logistics Ship 9
Average Speed (knots)	70	60	55	55	55	40	50
Full Performance Wave Height (m)	4	4	4	4	4	2.4	2.4
Range (nm)	5,000	5,000	10,000	8,700	8,700	800	1,000
Payload (mt)	4,537	11,797	11,797	4,537	6,806	1,312	726
Ramp Requirements	y	y	y	y	y	y	y
Total Crew	30	30	30	30	30	20	20
Structural Technology	far	far	far	far	far	near	near
Waterjet Technology	far	far	far	far	far	near	near
Prime Mover Technology	far	far	far	far	far	near	near

The intention was to cover as many vessel types as could reasonably be considered contenders for each mission. Monohull, catamaran, trimaran and Surface Effect Ship (SES) designs were produced for each of the missions. Other high-performance ship concepts were not included either because the speed/range/payload parameters of interest were judged to be incompatible with the performance attributes of those concepts or because the available technology base would not support development of the required designs.

Development of point designs for each of the hullform types to a uniform standard was a priority. Common design standards, margins, manning assumptions, and weight algorithms were adopted where practical and appropriate. A common philosophy for loading, stowing, and unloading cargo was used. In particular, technology projections from the HSS Technology Workshop for structures and materials, gas turbines, reduction gears, and waterjets were combined with additional technical information to produce a common basis for these technologies in the designs. Representative ship concepts for each of the four hullform types are illustrated in Figures 1-3 to 1-6. The overall proportions of the different ship types, arrangement of cargo spaces, and machinery plant concept are evident from the figures.

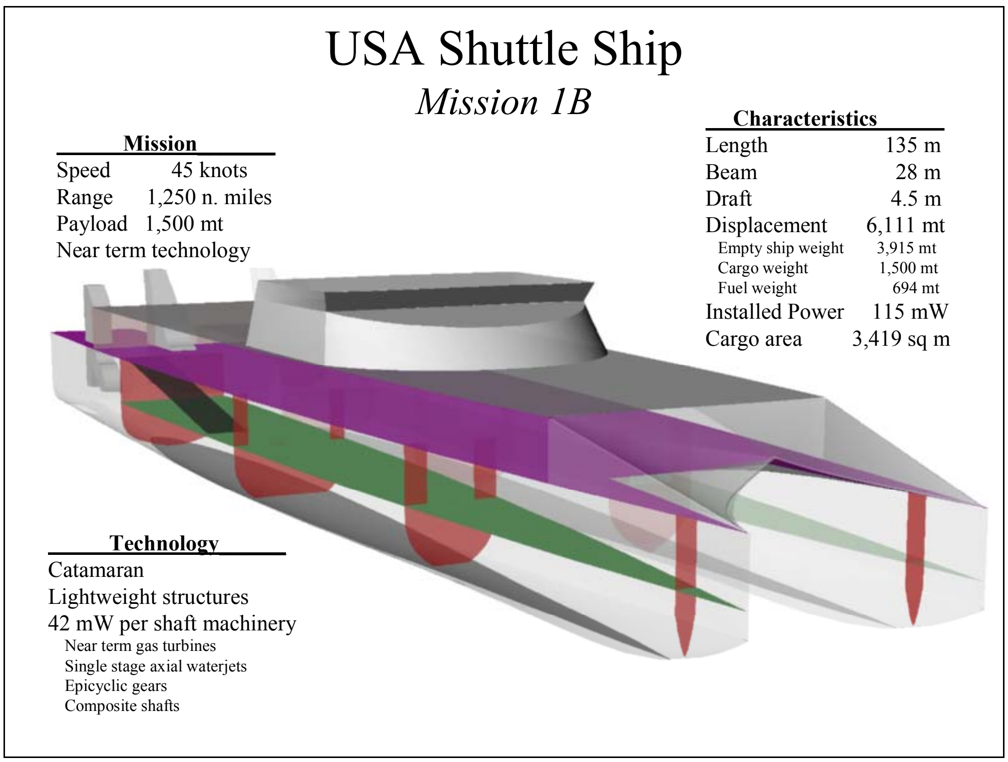


Figure 1-3: Representative Near-Term Technology Intra-Theater Catamaran

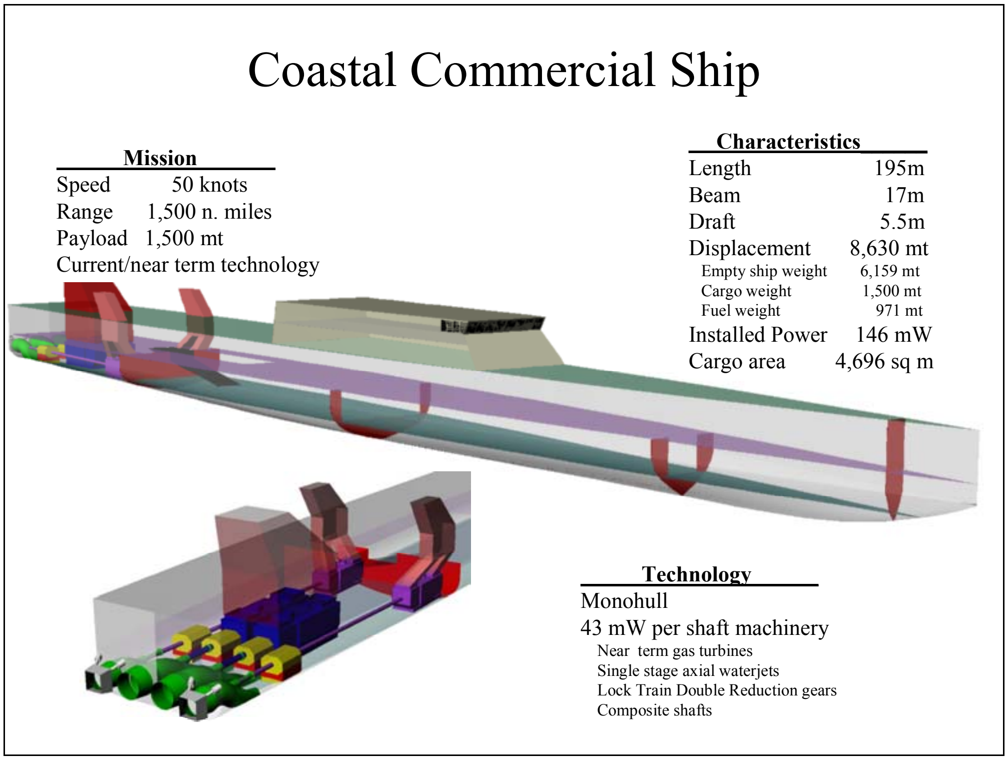


Figure 1-4: Representative Near-Term Technology Intra-Theater Monohull

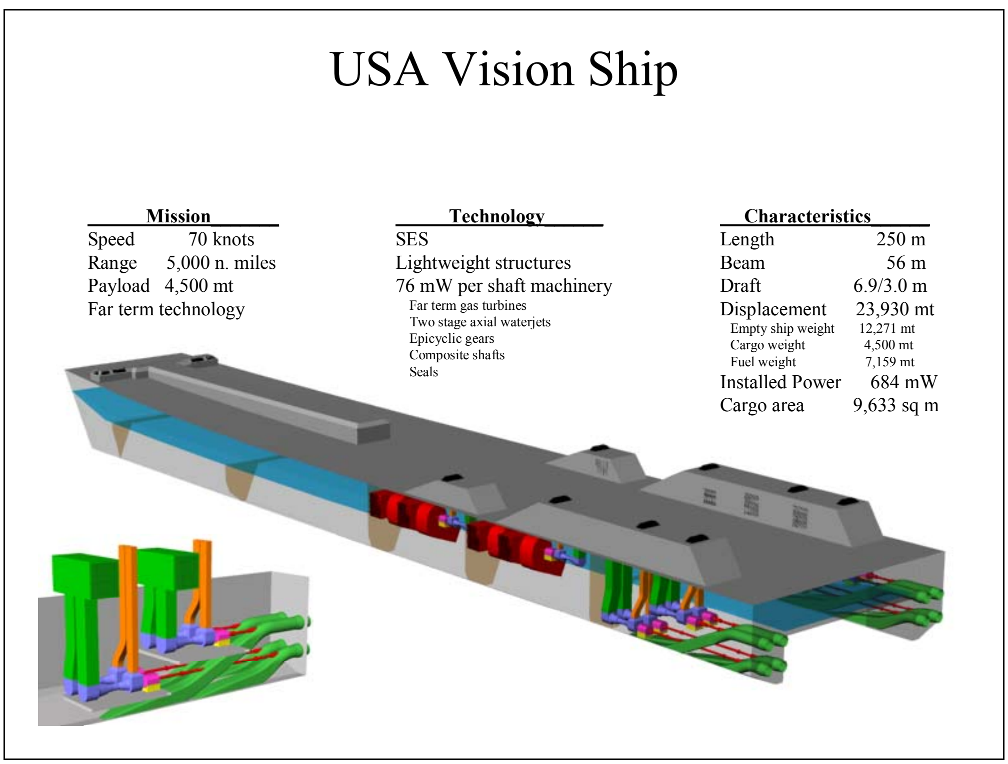


Figure 1-5: Representative Far-Term Technology Inter-Theater Surface Effect Ship

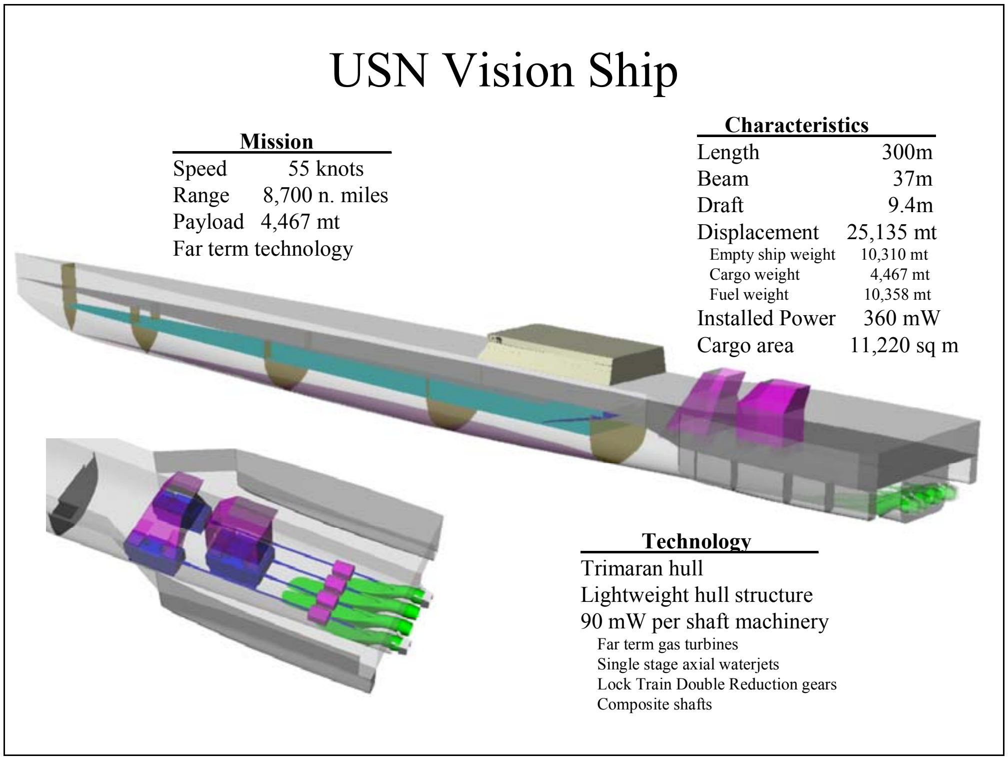


Figure 1-6: Representative Far-Term Technology Inter-Theater Trimaran

The resulting designs varied from small intra-theater ships displacing a few thousand tonnes to inter-theater ships with displacements in excess of 50,000 tonnes. Wide variations in the amount of installed power resulted from this size variation and the speeds required. Ship displacement and installed power for the designs are summarized in Figures 1-7 and 1-8. These designs are the basis for this technology development plan.

The capabilities needed from each of the technologies to produce these designs were compared with the technical state-of-the-art for those technologies to define the necessary near-term and far-term technology enhancements. Estimates of the time to develop and rough-order-of-magnitude development costs were made for each of the technologies based on a variety of factors including experience with development of similar technologies, engineering estimates, vendor data, and cost models. The goal of these plans is to bring the individual technologies to a level of maturity sufficient to lower risk to levels appropriate to ship design and construction. Technology development plans for each of the technologies are provided in the following sections of this report.

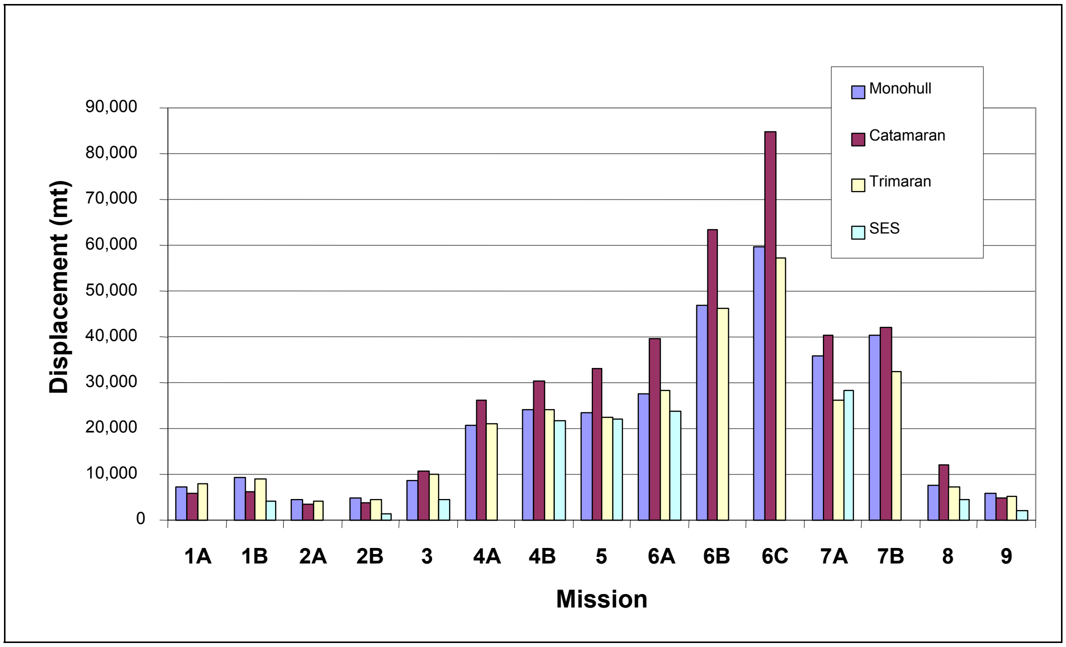


Figure 1-7: Full-Load Displacement of HSS Designs

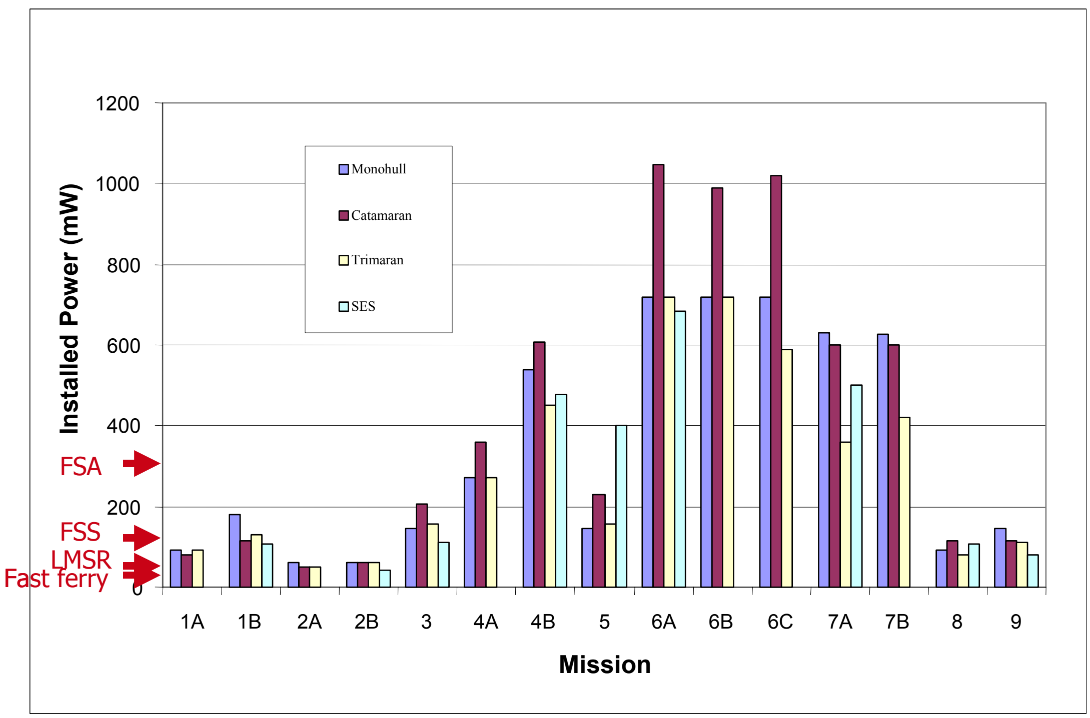


Figure 1-8: Installed Power of HSS Designs

These development plans are comprehensive with no allowance for market-driven technology development that may occur through commercial initiatives. Some technology development in critical areas is expected to meet anticipated commercial needs. For example, development of large gas turbine technology is highly likely for aerospace, industrial, and commercial marine projects. While such commercial technology development efforts will potentially reduce the need for Government investment, elimination of this investment is not expected since there is some risk that the commercial efforts will either not come to fruition or the commercially-derived capabilities will fall short of the capabilities needed to meet the more demanding military missions. Consequently, the potential existence of these commercial efforts is identified while the cost reductions that might result have not been shown.

The plans that follow contain some necessary redundancies since the specific need for some of the technologies depends on other technology choices. The choice of hullform technology has a particularly large impact on requirements for other technologies. For example, development of far-term SES hulls requires development of SES-peculiar lift fan and seal technologies. Alternately, monohull and trimaran hulls require development of locked train, double reduction (LTDR) gear technology, SES hulls require epicyclic reduction gear technology development, and catamarans require a mix of epicyclic and LTDR technologies. Since choices such as these cannot be made with certainty prior to commitment to specific long-term objectives, the redundancies have been identified and retained at the individual technology level. However, it is unlikely that the full matrix of technologies will be developed. Consequently, a representative comprehensive program is summarized in the last section of this report.

References

1. Ritter, O. and Templeman, M., "High-Speed Sealift Technology, Volume 1", Carderock Division, Naval Surface Warfare Center Technology Projection Report CDNSWC-TSSD-98-009, September 1998 (see http://www.ccdott.org).

2. Kennell C., D.R. Lavis, and M.T. Templeman, "High-Speed Sealift Technology", *Marine Technology*, Volume 3, No. 3, July 1998.

2.0 SHIP HULLFORM CONCEPTS

2.1 Introduction

This High-Speed Sealift Technology Development Plan is based on design studies produced using three displacement hullforms (monohull, catamaran, and trimaran) and one powered lift hullform (SES). All of these hullforms are considered viable candidates for HSS missions due to the existence of a technology base that is suitable to support design and construction of each hull type, although for lower levels of performance. Proof of concept demonstrations of each of these concepts has also been achieved through operational experience with large-scale ships including numerous commercial variants on these hulls, again for lower levels of performance. The higher levels of performance required for HSS missions result in hulls that are typically much more slender than existing variants. The increased slenderness of these hull concepts requires extrapolation beyond current capabilities in critical areas such as structural loads, resistance and powering, and seakeeping. The necessary technology development encompasses model test data, development of analysis tools, and development of design standards and practices such as those required for structural classification. While the technology extrapolations differ for the different hullforms, the magnitude of the extension is comparable for each.

Many of the far-term HSS designs produced are faster and significantly larger in size than similar existing ships. Several of these missions result in displacements above 20,000 tonnes. By comparison, most high-speed displacement ships, such as today's fast ferries, displace less than 2,000 tonnes. The largest SES displaces about 1,500 tonnes. The technical risk in extrapolating the current hulls to meet the more demanding HSS missions will be reduced significantly through design, construction, and technical validation of intermediate-size high-speed ships which use slender hulls similar to those envisioned for HSS roles. While such a progressive approach to evolution of hullform technology is prohibitively expensive if attempted for all of the hulls, it is strongly recommended for any hullform(s) chosen for development.

Results from design studies have shown that monohull, catamaran, trimaran, and SES hullforms are viable alternatives for HSS missions. However, monohull, trimaran, and SES variants were shown to offer superior weight, power, and fuel consumption advantages for missions resulting in displacements exceeding 10,000 tonnes. Most of these missions are inter-theater, have speeds of 50 knots or higher, and rely on far-term technology. In contrast, catamaran hulls were found to be attractive options for the smaller sizes. These intra-theater missions required near-term technology and typically had speeds below 50 knots. The existence of a mature catamaran industry that is producing commercial high-speed ferries with similar characteristics (2,000 tonnes displacement, 40 knots) indicates that only modest technology evolution is necessary to produce catamaran designs for these intra-theater missions.

2.2 Monohull

2.2.1 State-of-the-Art

While the U.S. shipbuilding industry has extensive experience designing and building monohulls of the size required for the HSS missions, speeds of these ships are generally much slower than

the 40-70 knots required for HSS designs. An extensive technology base has been developed over many decades to support reliable development of these slower traditional monohulls. This capability has been demonstrated for displacements in excess of 40,000 mt and speeds approaching 40 knots (SS United States). Even higher speeds have been demonstrated with special purpose designs such as the 1,100 mt yacht Destriero (54-knot Atlantic crossing), although the size of this ship is inadequate to support near or far-term HSS missions.

Achieving the speed and range requirements identified for near and far-term monohulls requires hulls that are much more slender than hulls used on traditional designs. Slenderness parameters of these advanced hulls are well beyond the existing technology base. As a result, expansion of the technology base is required to allow reliable prediction of vital design characteristics such as sea induced loads, resistance, powering, seakeeping, and maneuvering. The hydrodynamic integration of high-power waterjets into these slender hulls is of particular importance to minimize installed power, minimize fuel consumption, and assure reliable operation in representative sea conditions. The needed technology includes extension of analytic models and computer programs to address the slender hulls and higher speeds as well as comprehensive model test data.

Slenderness and high speed also have pronounced effects on structural design and performance. Hull girders for the hydrodynamically slender hulls are also structurally slender. Sensitivity of powering performance to weight, coupled with the magnitude of structural weight fractions for HSS designs, results in low structural weight being a design priority. Furthermore, high speeds are expected to result in significant slam loads in realistic seas. Consequently, structural loads and reactions to the loads such as slam induced whipping, both vertical and lateral, are expected to be of critical importance for slender high-speed monohulls. The resulting high-frequency, large-amplitude accelerations are expected to have significant effects on cargo, crew, and hull fatigue life.

The size-speed relationship of HSS far-term monohulls is compared with representative conventional ships in Figure 2.2.1-1. The figure illustrates the significant increase in speed required for these far-term HSS missions. While speed requirements for most near-term missions are much closer to demonstrated capability, slender hulls are also important for these slower designs to reduce installed power and fuel consumption.

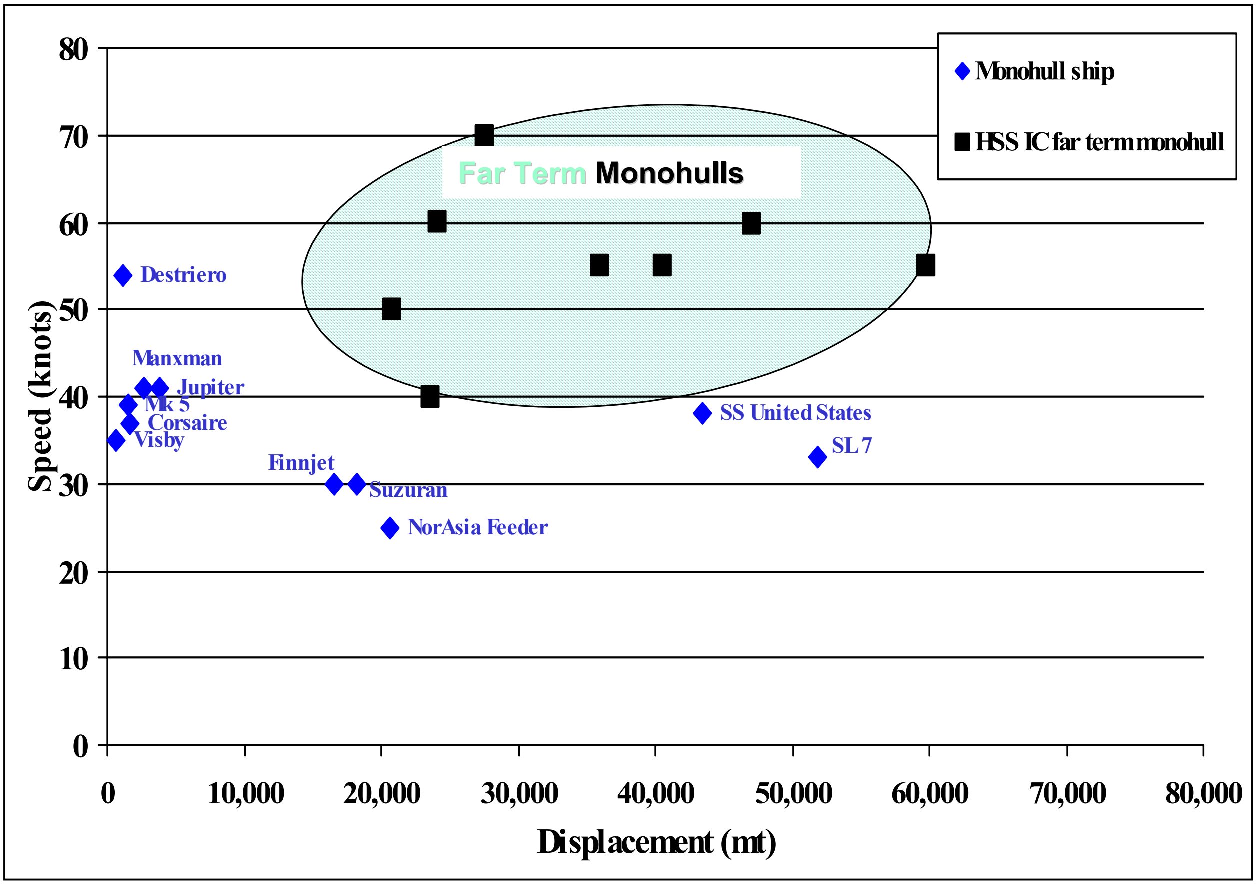


Figure 2.2.1-1: Monohull Technology

2.2.2 Technology Goals

Technology advances for slender high-speed monohulls are needed to reduce the risk associated with scaling small designs or models to the large displacements needed to support HSS missions. Technology development is required in the following areas:

> Structural loads – determination of the hydrodynamic forces (primary loads and slamming) and other loads that must be resisted by hull structure (covered under section 4.2 Loads).
>
> Resistance and powering – determination of total resistance due to friction, wavemaking, form drag, etc., added resistance in waves, and the total installed power required to attain a specified speed in specified sea conditions (covered under section 3.2 Powering).
>
> Propulsion – development of waterjet propulsors to provide the thrust needed to attain required speeds (covered under section 3.2 Powering and section 5.3 Waterjets).

Hull/propulsor integration – hydrodynamic integration of waterjets and hulls to minimize power and assure reliable seaway performance (covered under section 3.2.3 Hull/Propulsor Interaction).

Seakeeping – analysis of seaway-induced ship motions and their effect on ship and crew performance (covered under section 3.3 Seakeeping).

Maneuvering, dynamic stability, and control – analysis of turning capability, stability in turns, and dynamic control at high speed (covered under sections 3.4 Maneuvering and 3.5 Stability).

2.2.3 Overview of Development Plan

Technology development will be required to characterize the structural loads and hydrodynamic performance of large slender monohulls operating at high speed in rough water. Test data will be used to extend and validate analytical design tools and predictive methods, support development of classification standards, and increase confidence in the capability to produce successful designs of these large monohulls. Technology development efforts will focus on the development, analysis, and testing of representative slender monohull concepts selected to bridge the gap between the hullforms in the current technology base and HSS hulls. The tasks, time to complete each task, and cost associated with developing the needed monohull technology are shown in Figure 2.2.3-1. Two stages of hullform development, model testing, and analysis are shown to address variations in hullform expected and evolution of advanced hullform concepts. Costs shown are engineering estimates, based on expected scope of testing and facilities required. This hullform specific program will provide essential data to other technology development efforts such as powering (section 3.2), seakeeping (section 3.3), maneuvering (section 3.4), stability (section 3.5), loads (section 4.2), structural concepts (section 4.4), ABS HSS Guide (section 4.5), and waterjets (section 5.3). Similarity between monohulls and trimaran centerhulls will result in technology developed being applicable to both hull types.

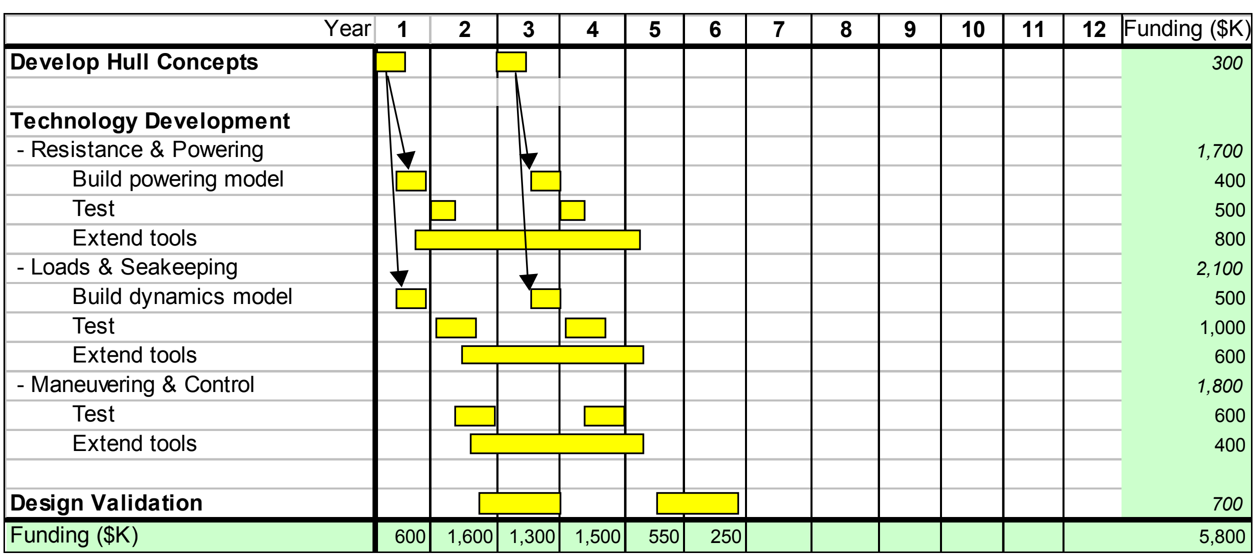


Figure 2.2.3-1: Monohull Technology Development Plan

2.3 Catamaran

2.3.1 State-of-the-Art

High-speed aluminum catamarans are widely used as vehicle and passenger ferries. Many designs are in service with displacements ranging from a few hundred tonnes to about 2,000 tonnes with speeds of 35-40 knots. Some small ferries have pushed the speed envelop above 50 knots, although generally only in sheltered waters. The largest aluminum catamaran, Stena's HSS 1500 (a special purpose semi-SWATH design), displaces 4,000 tonnes and makes 40 knots fully loaded. Range in commercial service of these high-performance ferries is generally a few (200-400) hundred miles. Virtually all of these ships have been designed and built outside the United States. The largest North American-built high-speed catamarans are the three 1,800-tonne, 34-knot Pacificat ferries recently built in Canada for Washington state.

The size-speed relationship of HSS near-term catamarans is compared with representative conventional ships and high-speed ferries in Figure 2.3.1-1. The figure shows that only modest increases in speed and ship size are required for near-term HSS missions. The larger, faster far-term catamaran designs are not shown since design studies have shown displacement, installed power, and fuel consumption of these large ships to be much greater than for other (monohull, trimaran, SES) HSS hulls for these high-speed, long-range missions. Catamaran hulls for these high-speed, long-range missions were found to be uncompetitive.

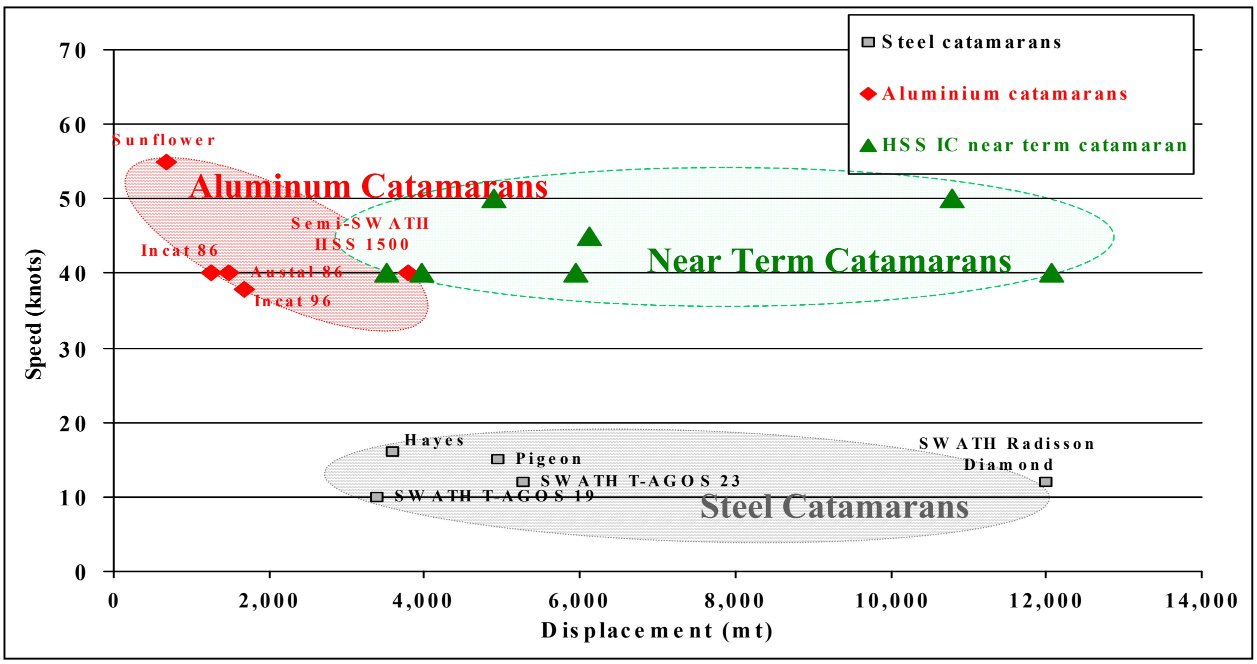


Figure 2.3.1-1: Catamaran Technology

The U.S. shipbuilding industry has very limited experience designing and building catamarans of the size and speed required for HSS missions. A number of small passenger ferries have been

built with speeds below 35 knots. These vessels have generally been built under license to foreign designs. Large steel catamarans displacing 3-5,000 tonnes with speeds below 20 knots have also been designed and built domestically during the 1960s for Navy missions. Significant technology was developed for these slow, open-ocean ships addressing critical issues such as powering, seakeeping, maneuvering, loads, and structural design. More recently, two classes of SWATH ships (T-AGOS 19 and T-AGOS 23), a specialized variant of the catamaran form, were also built for Navy missions. This domestic experience, the existence of a mature international high-speed catamaran industry, and the existence of partnering agreements between U.S. shipyards and foreign catamaran designers/builders results in assured availability of the catamaran technology needed to build near-term catamarans. Resolution of remaining technical issues such as development of designs to ABS High-Speed Craft Rules at the sizes of interest, completion of training and technology transfer efforts between foreign builders and their U.S. partners, and adaptation of DNV High-Speed Light Craft Rules-based high-speed ferry designs to meet the more stringent military requirements should result from ongoing commercial development. Consequently, investment in catamaran technology is not recommended. Although hullform-specific technology development is not recommended, future catamarans will benefit from the more generic technology development in structures and materials, gas turbines, reduction gears, and waterjets. In addition, the seakeeping and maneuvering tool extensions required for trimarans (see sections 3.3.2 and 3.4.2) will be produced in a generic multihull manner that will also allow modeling and analysis of catamarans.

2.3.2 Technology Goals N/A

2.3.3 Overview of Development Plan N/A

2.4 <u>Trimaran</u>

2.4.1 State-of-the-Art

The U.S. shipbuilding industry has limited experience designing and building trimarans of the size and speed required for the HSS missions. A few small, slow prototypes have been built as pleasure craft. Technology from model tests, full-scale trials, and design analysis has been produced under the UK/US trimaran joint trials program for the 1,200-tonne, 20-knot trimaran RV Triton. Limited design and model test experience with hulls similar to HSS trimarans has also resulted from commercial efforts such as those of Kvaerner Masa Marine and Nigel Gee & Associates which have explored a range of concepts including hulls displacing over 20,000 tonnes with speeds above 60 knots.

The HSS trimaran hullforms produced are essentially slender monohulls with very small sidehulls added to provide buoyant stabilization. HSS trimaran sidehulls typically provide only 2% of total buoyancy. While the sidehulls add complexity, most technical aspects of trimaran centerhulls may be viewed as essentially indistinguishable from the slender monohulls discussed in section 2.2. Consequently, the extensive monohull technology base is also applicable to trimaran centerhulls. Similarly, the technology extensions resulting from increased slenderness of HSS monohulls are also required for trimarans. Additional trimaran specific extensions are required to address sidehull related issues such as resistance, flow characteristics, seakeeping,

loads, structural response, and maneuvering and control of the combined sidehull/mainhull. While these trimaran-specific technology requirements add complexity, the overall effort is of the same scope and magnitude as that of the monohull.

The size-speed relationship of HSS far-term trimarans and monohulls is compared with representative conventional ships in Figure 2.4.1-1. As with the monohull case, a significant increase in speed is required for these far-term HSS missions. Speed requirements for most near-term missions are much closer to demonstrated capability.

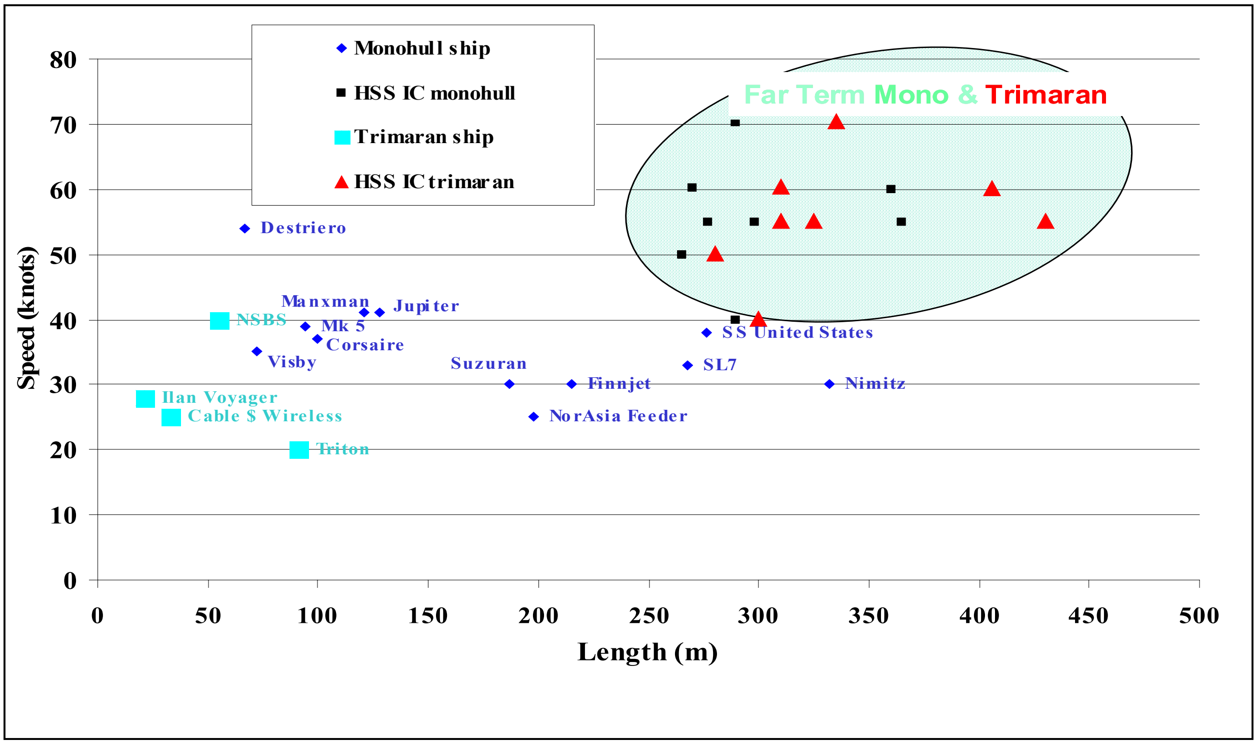


Figure 2.4.1-1: Trimaran Technology

2.4.2 Technology Goals

Technology advances for slender high-speed trimarans are needed to reduce the risk associated with scaling small designs or models to the large displacements needed to support HSS missions. Technology development is required in the following areas:

> <u>Structural loads</u> – determination of the hydrodynamic forces (primary loads and slamming) and other loads that must be resisted by hull structure (centerhull, sidehull, and cross-structure) (covered under section 4.2 Loads).
>
> <u>Resistance and powering</u> – determination of total resistance due to friction, wavemaking, form drag, etc., added resistance in waves, and the total installed power required to attain a specified speed in specified sea conditions (covered under section 3.2 Powering).

Propulsion – development of waterjet propulsors to provide the thrust needed to attain required speeds (covered under section 5.3 Waterjets).

Hull/propulsor integration – hydrodynamic integration of waterjets and hulls to minimize power and assure reliable seaway performance (covered under section 3.2.3 Hull/ Propulsor Interaction).

Seakeeping – analysis of seaway-induced ship motions and their effect on ship and crew performance (covered under section 3.3 Seakeeping).

Maneuvering, dynamic stability, and control – analysis of turning capability, stability in turns, and dynamic control at high speed (covered under sections 3.4 Maneuvering and 3.5 Stability).

2.4.3 Overview of Development Plan

Technology development will be required to characterize the structural loads and performance of large slender trimarans operating at high speed in rough water. Test data will be used to extend and validate analytical design tools and predictive methods, support development of classification standards, and increase confidence in the capability to produce successful designs of these large trimarans. Technology development efforts will focus on the development, analysis, and testing of representative slender trimaran concepts selected to bridge the gap between the hullforms in the current technology base and HSS hulls. The tasks, time to complete each task, and cost associated with developing the needed trimaran technology are shown in Figure 2.4.3-1. Two stages of hullform development, model testing, and analysis are shown to address expected variations in hullform and evolution of hullform concepts. Costs shown are engineering estimates, based on expected scope of testing and facilities required. This hullform specific program will provide essential data to other technology development efforts such as powering (section 3.2), seakeeping (section 3.3), maneuvering (section 3.4), stability (section 3.5), loads (section 4.2), structural concepts (section 4.4), ABS HSS Guide (section 4.5), and waterjets (section 5.3). Similarity between trimaran centerhulls and monohulls will result in technology developed being applicable to both hull types.

Year	1	2	3	4	5	6	7	8	9	10	11	12	Funding ($K)
Develop Hull Concepts													*300*
Technology Development													
- Resistance & Powering													*1,700*
Build powering model													400
Test													500
Extend tools													800
- Loads & Seakeeping													*2,700*
Build dynamics model													500
Test													1,000
Extend tools													1,200
- Maneuvering & Control													*1,800*
Test													1,000
Extend tools													800
Design Validation													*700*
Funding ($K)	600	2,050	1,550	1,950	800	250							7,200

Figure 2.4.3-1: Trimaran Technology Development Plan

2.5 <u>SES</u>

2.5.1 State-of-the-Art

The SES has approximately 40 years of developmental and operational experience in the U.S. and abroad. The U.S. Navy had, at one time, completed a design and intended to construct a high-speed (80-knot), transoceanic, 3,000-ton low length/beam (L/B) ratio SES (3KSES). This aggressive acquisition program evolved from a technology base that included model tests, analysis, and operation and testing of a series of small manned test craft. During the program, two 100-ton test craft, SES 100A and SES 100B, were built and evaluated in trials at speeds approaching 90 knots to reduce program risk. While the 3KSES program was terminated prior to the construction phase in 1979, a firm SES technology base resulted from the effort. The air-cushion vehicle (ACV) is a related technology that has paralleled SES development and has many common technology areas. Subsequently, the U.S. Navy operated what was the largest known SES through the 1980s, the 200-ton, 40-knot SES-200. The L/B of this craft is somewhat higher than that of the higher speed craft. The U.S. Coast Guard also operated the smaller 152-tonne, 30-knot SES 110 'Seabird' class. Variants of the SES 110 design were also built as commercial crew boats and as a hydrographic survey boat. Closely related technology was also developed as part of air cushion vehicle (ACV) programs, exemplified by the U.S. Navy's Landing Craft, Air Cushion (LCAC), of which 91 have been built.

Significant development of SES technology has occurred as a result of international programs as well. In 1990, the Soviet Union commissioned the largest SES to that time, the 1,000-ton Dergach. The SES size boundary was extended again in 1994 when the 54-knot Japanese Techno-Superliner TSL-A70 was built with a displacement of 1,500 tons.

A 700-ton SES test craft underwent detailed development in the Federal Republic of Germany in cooperation with the U.S. However, the project was cancelled before construction began. While

this experience augments the U.S. technology base, a significant jump in technology is needed to bridge the gap between these ships with displacements below 1,500 tons and the 20,000^{+} ton HSS SES concepts as shown in Figure 2.5.1-1.

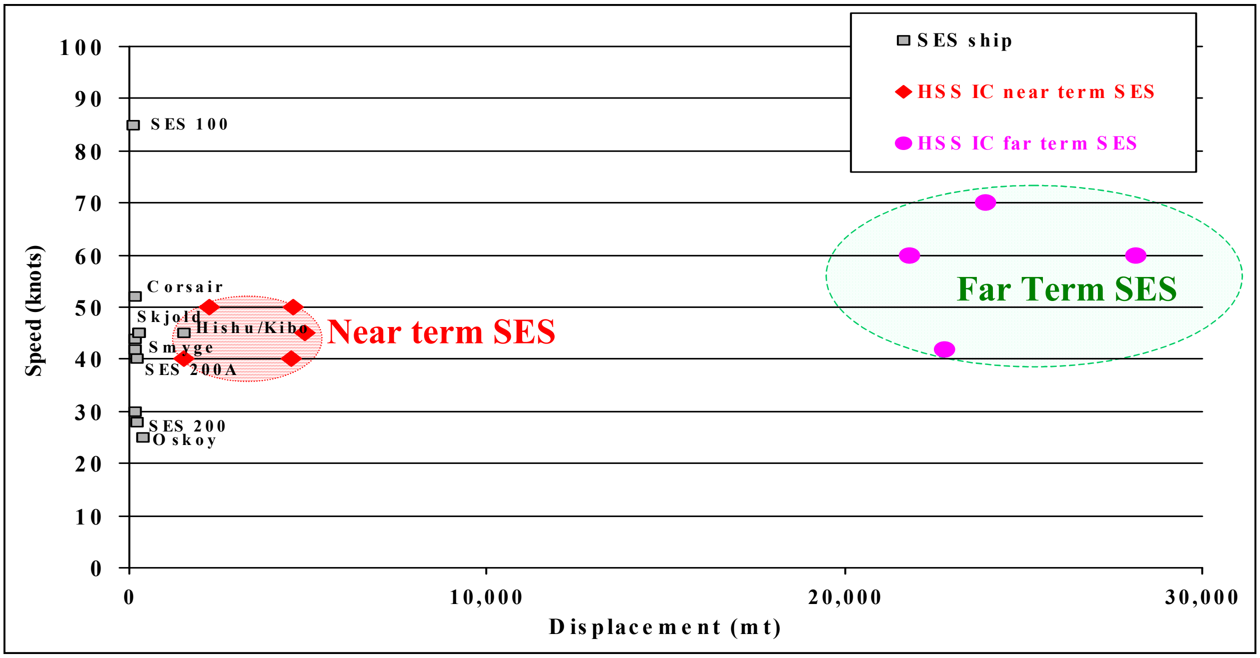


Figure 2.5.1-1: SES Technology

2.5.2 Technology Goals

Technology advances for high-speed, high-L/B SES hulls are needed to reduce the risk associated with scaling small designs or models to the large displacements needed to support HSS missions. Technology development is required in the following areas:

> Structural loads – determination of the hydrodynamic forces (primary loads and slamming) and other loads that must be resisted by hull structure (covered under section 4.2 Loads).
>
> Resistance and powering – determination of total resistance due to friction, wavemaking, form drag, etc., added resistance in waves, and the total installed power required to attain a specified speed in specified sea conditions (covered under section 3.2 Powering).
>
> Propulsion – development of waterjet propulsors to provide the thrust needed to attain required speeds (covered under section 5.3 Waterjets).
>
> Hull/propulsor integration – hydrodynamic integration of waterjets and hulls to minimize power and assure reliable seaway performance (covered under section 3.2.3 Hull/Propulsor Interaction).

<u>Seakeeping</u> – analysis of seaway-induced ship motions and their effect on ship and crew performance (covered under section 3.3 Seakeeping).

<u>Maneuvering, dynamic stability, and control</u> – analysis of turning capability, stability in turns, and dynamic control at high speed (covered under sections 3.4 Maneuvering and 3.5 Stability).

<u>Seals</u> – development of high-performance, durable end seals and transverse seals needed to meet powering and seakeeping goals (covered under section 5.6 SES End Seals).

Additional SES-specific systems level technology development is addressed in section 5.5 SES Lift Fans and 5.7 Packaging.

2.5.3 Overview of Development Plan

Technology development will be required to characterize the structural loads and hydrodynamic performance of large, high-L/B SES operating at high speed in rough water. Test data will be used to extend and validate analytical design tools and predictive methods, support development of classification standards, and increase confidence in the capability to produce successful designs of these large SES. Technology development efforts will focus on the development, analysis, and testing of a representative high-L/B SES concept selected to bridge the gap between the hullforms in the current technology base and HSS hulls. The tasks, time to complete each task, and cost associated with developing the needed SES technology are shown in Figure 2.5.3-1. Costs shown are engineering estimates, based on expected scope of testing and facilities required. This hullform specific program will provide essential data to other technology development efforts such as powering (section 3.2), seakeeping (section 3.3), maneuvering (section 3.4), stability (section 3.5), loads (section 4.2), structural concepts (section 4.4), ABS HSS Guide (section 4.5), waterjets (section 5.3), lift fans (section 5.5), and seals (section 5.6).

Time to complete (years)	1	2	3	4	5	6	7	8	9	10	Est. Cost ($K)
Technology Development											
- Resistance & Powering											400
- Loads											800
- Maneuvering & Dynamic Stability											600
- Seakeeping											300
Funding ($K)	1,600	500									2,100

Figure 2.5.3-1: SES Technology Development Plan

3.0 HYDRODYNAMICS

3.1 Introduction

HSS mission speed and range objectives can be met using very slender displacement hulls and high length-to-beam (L/B) ratio surface effect ships. The increased slenderness of these hull concepts requires extension of hydrodynamic technology in critical areas such as structural loads, resistance and powering, and seakeeping. The necessary technology development encompasses model test data, development of analysis tools, and development of design standards and practices such as those required for structural classification. While the technology extrapolations differ for the different hullforms, the magnitude of the extension is comparable for each.

3.2 Powering

The principal emphasis of all HSS missions is speed. The importance of speed is magnified by the weight implications of machinery and the fuel needed for the ranges required, particularly for the inter-theater missions. It is, therefore, of critical importance that speed and power predictions be accurate for HSS hulls in calm water and waves. Of critical importance is the need to understand the flow about these slender hulls to allow effective integration of large high-power waterjets.

The purpose of this effort is to extend resistance and powering prediction techniques to address the slender hulls needed and provide a validated basis for sizing and selecting appropriate propulsion systems. A major objective is to validate analytic models as design tools to support development of slender high-speed configurations.

The approach to be used to develop powering technology will be based on the following:

- develop hull designs that meet representative requirements using existing data and state-of-the-art analytical tools (e.g. Computation Fluid Dynamics methods).
- predict ship resistance and powering performance and flow about the hulls with appropriate analytic and empirical tools.
- plan and conduct tow tank tests to verify predictions. The models will be designed to represent hull geometries and waterjet propulsors appropriate to HSS missions and will be tested for a range of operating conditions, speeds, and sea states.
- correlate test data with predictions to extend and validate predictive techniques.

3.2.1 State-of-the-Art

Resistance estimates for the HSS monohull, catamaran, and trimaran designs rely heavily on systematic series model test data such as Series 64 and Taylor Standard Series. Hullform geometry of HSS hulls ($L/\nabla^{1/3}$, section shapes, transom size, bow shape, etc.) differs markedly from the hulls in these standard series. For example, hulls for traditional high-speed ships (e.g. SS United States, SL 7, aircraft carriers, surface warships) and large cargo ships (LMSR, T-AKR

287, T-AKR 5069) have slenderness values below $L/\nabla^{1/3} = 8.0$. By comparison, HSS displacement hulls range between $10 < L/\nabla^{1/3} < 12$. Such differences have significant effect on hull resistance, hull/propulsor integration, and powering requirements for high-speed displacement ships. While limited proprietary data exists in the form of model test data and design data associated with development of a few commercial concepts, a comprehensive database to support development of these much more slender hulls is not publicly available.

The absence of appropriate systematic series data has forced designers to resort to advanced computational techniques (CFD) to address critical powering needs such as resistance of unusual hulls and flow characteristics (bulbs, waterjet inlets, transoms, streamlines over hulls). More challenging is the need to model the flow about a hull with operating high-power waterjets, an essential step toward optimal hull/propulsor integration for peak power and fuel efficiency. However, these analytical methods require careful correlation with physical data to assure accuracy. This test data is not readily available for slender HSS displacement hulls at the high speeds of interest. Absence of this data is a severe obstacle to the development of mission-specific designs and also hinders generic high-speed hull research and design tool development.

Tools to predict SES resistance and powering requirements have been developed over the past forty years through a combination of model tests, manned test-craft trials and analytical models. The U.S. Navy's SES study programs began in the early 1960s and have progressed with a series of dedicated test-craft development programs. Through 1979, the major thrust of the SES effort was directed towards high speed (60-100 knots), low length-to-beam (L/B) ratio SES. Following the termination of the 3KSES program in 1979, the U.S. Navy redirected the SES studies to higher L/B ratios and slower speeds (e.g. 25-55 knots). Higher L/B ratio SES are considered more practical in terms of structural and powering requirements for very large ships such as the HSS mission ships. The fundamental question that needs resolution is the ability of analytical models to predict SES performance with sufficient accuracy to support the design of large ($25{,}000^{+}$ tonne) vessels. Application of CFD tools to hull/propulsor integration is as challenging for SES hulls as for displacement hulls. Consequently, a similar need exists for test data for high-L/B SES hulls at the high speeds of interest.

3.2.2 Technology Goals

The objective of the monohull, trimaran, and SES powering work is to develop a comprehensive technology base for slender high-speed HSS hulls and validate analytical techniques for prediction of full-scale resistance and powering for HSS hulls. The following approach will be used:

- Review and analyze existing data to define extensions to analytical models needed for HSS hulls.
- Modify and update analytical models.
- Modify and update test techniques for high-speed hulls.
- Conduct comprehensive model tests to produce data to validate HSS hulls and analytical predictions.

- Utilize data from model tests and operational ships to validate predictive techniques by correlation.

Comprehensive tests and analyses will be used to extend analytical methods and validate computer models for slender HSS displacement hulls ($10<L/\nabla^{1/3}<12$) and high-L/D (L/D~6) SES hulls. The approach is to expand current databases and to build upon proven existing analytical models.

Extension of seal technology is required for large HSS SES hulls. An approach that combines model testing and analytic methods is needed to develop seal systems for SES HSS missions. Seals with long life, low resistance, and good motions characteristics are needed. The potential performance enhancement using mid-cushion transverse seal concepts will be assessed.

3.2.3 Hull/Propulsor Integration

Current practice for designing waterjet-propelled hulls is to first design a hull with low drag followed by the design of waterjets with good propulsive efficiency. Waterjet influence on the hull design is minimal, consisting primarily of geometric requirements for the fit of the machinery and inlets. Conversely, waterjet design is influenced by the hull. Flow irregularities in the waterjet inlet are major factors in the design of waterjet components such as inlet ducts, stators, and rotors.

Omitted from the hull design process are the changes in hull flow properties resulting from operation of the waterjets. These changes result from alteration of the pressure distribution near the stern caused by waterjet inlet suction under the hull and exhaust behind the transom. Resistance, sinkage, trim, and the direction of the streamlines over the hull are affected. The draw-down of the water surface in the vicinity of the waterjet inlets is of particular concern since it increases the likelihood of air injestion by the waterjets in a seaway. While pertinent to the design of all waterjet-powered designs, the importance of these flow changes is magnified by the slender hulls and high installed power of HSS concepts. Potential consequences of this lack of integration include reduced efficiency of the waterjet, higher fuel consumption, and operational limitations in waves.

Extension of existing design tools, including Computational Fluid Dynamics (CFD) techniques as well as the model test techniques needed to validate predictions, is a goal of this plan.

3.2.4 Overview of Development Plan

The displacement hull technology development effort will extend the existing technology base to encompass the more slender hulls of HSS monohulls and trimarans. The needed extensions will be produced using advanced analytic methods, model test data, and available full-scale data. A major objective is to validate analytic models as design tools to support development of slender high-speed configurations.

Representative hulls with $L/\nabla^{1/3}$ of 10-12 and waterjets will be developed using available analytic and empirical data. Two hull concepts will be developed to address variations in

slenderness and secondary hull characteristics such as section shape, beam/draft ratio, etc. Resistance, powering, sinkage, trim, and flow data will be measured at model-scale for these hulls to assess resistance characteristics, hull/propulsor interactions, and flow properties such as streamlines on the hulls, flow in the inlets, transom flow, and pressure distribution on the hulls. Comparisons between measured data, estimates produced to develop the modeled hulls, and post-test analysis will be used to establish credibility of the design tools, identify and eliminate shortfalls in the technology, and validate performance of HSS hulls. Validation will be further enhanced using data for similar slender hulls developed by commercial projects where available. The following process will be followed in this high-$L/\nabla^{1/3}$ displacement hull powering effort:

1. Develop hullform, inlets, and propulsion-system design for a high-$L/\nabla^{1/3}$ HSS displacement hull using analytical methods, model test data, and full-scale data.
2. Prepare a model test plan to verify resistance, inlet performance, powering, and performance.
3. Design and fabricate scale models of the hulls and propulsors.
4. Conduct model tests and reduce data.
5. Analyze test data and correlate with performance predictions.

Waterjet inlet simulation tests will be conducted to assess the inlet design, arrangement of the pumps, and hull/propulsor integration. The effects of operation of individual waterjets and combinations of waterjets will be assessed.

The scheduling and costing plan for resistance and powering technology development is shown in Figure 2.2.3-1 for monohulls and Figure 2.4.3-1 for trimarans. The scheduling and costing plan for hull/propulsor integration technology development is shown in Figure 3.2.4-1 for monohulls and trimarans.

Time to complete (years)	1	2	3	4	5	6	7	8	9	10	Funding ($K)
CFD Analysis											
- Geometry definition	▬										400
- CFD Analysis		▬									1,000
Hull-Propulsor Tests											
- Monohull		▬	▬								1,250
- Trimaran		▬	▬								1,250
- SES			▬	▬							1,250
Design Methodology Validation			▬	▬							750
Funding ($K)	400	2,400	2,200	900							5,900

Figure 3.2.4-1: Hull/Propulsor Integration Technology Development Plan

The SES technology development effort will validate SES analytical design tools for high-L/B HSS hulls. L/B for these advanced hulls is ~ 6. The initial objective is to correlate analytic predictions, model test data, and full-scale trials data for the SES-200, the highest L/B SES (L/B~4) built to date. Model tests of the SES-200 will be conducted to generate the necessary

data. Additional, less meticulous comparisons will also be made using available data for other (lower L/B) SES such as the 1,500 tonne Japanese TSL-A, Norwegian MCM SES, and commercial SES. This will provide the most comprehensive database of model, full-scale and analytical predictions to validate the design tools in the absence of full-scale trials of a high-L/B SES.

Extension of this technology base to the higher L/B~6 HSS hulls requires additional model testing. A hullform, propulsion system, lift-air supply system, and seal system will be developed reflecting technology assumptions for the HSS SES. Model tests will be conducted to verify the performance within the design operational parameters. The purpose of these tests will be to verify the integrated performance of the hull, propulsion system, and lift system, and to provide sufficient data for a design database for large ($25,000^+$ ton), high-L/B SES. The following tasks will be performed to support this high-L/B SES effort:

1. Develop hullform, seal concept, propulsion-system design, and lift-system design for a high-L/B HSS SES using analytical methods, model test data, and full-scale data.
2. Prepare a model test plan to verify powering performance including suitability of transverse seals.
3. Design and fabricate a scale model of the hull, propulsors, and lift system including a method to isolate seal performance from sidehull effects.
4. Conduct model tests and reduce data.
5. Analyze test data and correlate with performance predictions.

A waterjet inlet simulation test will be conducted to assess the inlet design and arrangement of the pumps in each side-hull. The effects of operation of individual waterjets and combinations of waterjets will be assessed.

The scheduling and costing plan for SES resistance and powering technology development is shown in Figure 2.5.3-1. The scheduling and costing plan for hull/propulsor integration technology development is shown in Figure 3.2.4-1 for SES.

3.3 <u>Seakeeping</u>

Low-L/B SES with active ride-control systems installed have demonstrated good ride quality for sizes through 1500-ton. This experience has identified scaling issues that must be resolved to assess seakeeping performance (motions, ride quality) as well as design the lift system for high-L/B HSS size ships. Current seakeeping simulation and ride-control system technology has also provided considerable insight regarding SES cushion dynamics. The "bunching" of SES excitation and resonance frequencies in the seasickness range can lead to a variety of development difficulties. Cushion heave dynamics of a full-scale SES cannot be represented at model-scale due to an inability to scale atmospheric pressure. Consequently, development of design solutions is complicated by an inability to use model-scale motions directly. Current ride-control system analysis and design techniques have demonstrated viable approaches for predicting, evaluating, and controlling cushion dynamics problems up to SES-200 size ships. However, they must be refined and verified for the high-L/B, large HSS design application.

3.3.1 State-of-the-Art

A robust capability for evaluating seakeeping performance of monohull displacement hulls is currently available. Fundamental to this capability are frequency domain computer models based on thin-ship theory that assess the statistical properties of ship motions. Supporting this statistically-based frequency domain foundation are the more complex time-domain programs that predict actual motions of a ship in a specific wave system. These readily available and widely used programs are well validated for conventional monohulls. HSS displacement monohulls are more slender than conventional ships, incorporate different shapes, and operate at higher speeds (or Froude no. - Fn) than current monohulls. While these hullform features are expected to be compatible with current seakeeping tools, validation with test data for representative slender HSS hulls at Fn of interest will enhance credibility.

Seakeeping assessments of high-speed, slender displacement catamarans and trimarans are more complex than for monohulls. While the fundamental physics of multihull motions are the same as for monohulls, experience with multihulls such as SWATH ships and slow-speed conventional catamarans has shown that multihulls require significant extensions to monohull seakeeping technology to accurately model non-monohull features such as between-hull interactions, differences in damping, and above-water geometry. Furthermore, the more limited demand for multihull motions prediction capability has inhibited development of ship motions tools for these hulls. While prediction capability exists for catamarans, extension of the tools to more accurately model the hull geometry and hydrodynamic effects of high-$L/\nabla^{1/3}$ HSS displacement catamarans and trimarans is needed. Additional test data (section 2.4.3) is also needed for representative HSS hulls to guide and validate these extensions.

Prediction of seakeeping performance of SES is more complicated than for displacement hulls. While prediction capability exists, extension of this technology is needed due to the greater size and higher L/B of HSS SES hulls. SES seakeeping technology includes ship motions, seakeeping, ride quality (habitability), and overall air-cushion system dynamics. The ship systems involved or affected include the air-cushion, the lift system, the bow and stern seals, and the ride-control system (RCS). Hull design is also a consideration in that side-hull hydrodynamics make a major contribution to ship motions.

The subject of ship motions, ride quality, and cushion dynamics has long been an integral element of SES development in the U.S., dating back to the XR-1, XR-3 and SES-100 A&B design and test-craft programs of the late 1960s. The early test-craft design and development programs identified a number of key aspects of SES cushion system development, including:

1. SES motions differ from those of conventional ships due to the dynamic nature of the air-cushion suspension system, the higher frequencies of encounter, and the catamaran hull-form.
2. SES motions cannot be adequately scaled from tow-tank model test results due to the fact that an important factor in cushion dynamics is the ratio of absolute cushion pressure to atmospheric pressure. This ratio cannot be properly represented in model tests run at atmospheric pressure.

3. SES cushion systems are subject to a variety of acoustic and structural resonances and instabilities. These effects can limit available gain on an active ride-control system and amplify ship motions.

Since control of ambient air pressure and density in large tow-tank facilities is not economically feasible, SES development programs have adopted the concept of computer program simulation for all parameters that are affected by cushion compressibility. The motion simulation programs are applied to predict results of model tests and existing test-craft operations. If good agreement is achieved, the simulation programs are considered valid for prediction of cushion pressure variations and motions at full-scale.

Analysis and design efforts under the early test-craft programs recognized the unusual nature of SES motions and the need for active control systems to improve ride quality and habitability. The effect of SES motions on crew performance was quantified by applying simulated full-scale motions (with and without ride-control) to volunteer subjects for extended periods. These tests confirmed the need for SES ride-control to reduce crew fatigue and motion sickness. Ride-control systems based on controlled cushion venting were designed and installed in the early SES-100A and SES-100B test craft. The XR-1 was also eventually equipped with variable-flow fans.

Cushion resonance and instability problems resulted in severe gain limitations and reduced effectiveness for the initial XR-1, SES-100A and SES-100B ride-control systems. An extended program for improvement of ride-control effectiveness was undertaken in support of 2KSES/3KSES development using the XR-1 test craft for performance evaluation. This program was discontinued, however, after the cancellation of the 3KSES in 1979.

SES computer motion simulation programs were developed by Aerojet and Bell in support of the XR-3, SES-100A, SES-100B and 2KSES programs. Subsequently, these programs were developed further by Rohr Marine Inc. (RMI), Oceanics, and Maritime Dynamics Inc. (MDI) in support of the 3KSES program. These computer simulations fell into two categories:

1. Linearized, frequency-domain programs which predicted motion statistics but which had limited capability for predicting discrete events such as slamming and broaching (Aerojet, Rohr Marine, Maritime Dynamics).
2. 6-DOF time-domain programs representing discrete events but with limited capability for motion statistics (Aerojet, Wyle Labs, Payne Inc., Oceanics, Textron Marine).

The predictions of these programs were correlated against model test results, SES-100A and SES-100B test data, and particularly XR-1D test results.

Navy acquisition of the SES-200 test craft in 1981 provided an SES platform of substantially increased size and capability. A digital microprocessor-based RCS, driving deck-mounted vent valves, was installed on the SES-200 and an extensive test and evaluation program was conducted during 1983. Ship motions and ride quality data were acquired under a wide variety of operating conditions. The effectiveness demonstrated by the SES-200 ride-control system during these tests is illustrated by the fact that r.m.s. cushion pressure variations were reduced by

up to 60% and r.m.s. heave accelerations at the C.G. were reduced by nearly 50% under some conditions.

Maritime Dynamics, Incorporated (MDI) continued development of improved computer simulations for SES motions and cushion dynamics, and in the design and demonstration of more effective ride-control systems. An advanced digital RCS involving "distributed" vent valves and a multi-input/multi-output controller was installed in the SES-200 test craft.

The largest SES built to date, the TSL-A at 1,500 tons, uses a combination RCS involving both vent valves and T-foils.

Other than standard modern control system technology, the primary design tools applicable to the SES motions/ride quality/cushion dynamics area consist of digital computer programs for simulation of SES dynamics. These programs typically accept input which defines an SES of any size and hull configuration. In addition, input parameters, or "option" modules, allow selection of alternate lift systems, bow and stern seals, and ride-control systems.

A number of programs are currently available, including:

- SES 5-DOF Seakeeping Program (Maritime Dynamics, Inc.) – This design tool is a linearized, frequency-domain SES motion simulation. The program predicts the statistics of vertical plane and lateral plane motions for an SES operating in a random seaway specified by a wave energy spectrum. The rigid-body ship motions are limited to 5 degrees of freedom (no surge). The ship dynamics are linearized about mean operating conditions for each case (trim, draft, cushion fan flow, etc.) that are established by operator inputs. This program is primarily intended to evaluate SES motion statistics in a seaway, with or without ride-control. It has been refined and extended by MDI since it was first developed by Aerojet in 1969. Its predictions have been correlated and validated using tow-tank test data from several model programs and using the extensive SES-200 test and evaluation database.
- SES Finite-Volume Vertical Plane Motions Program (Maritime Dynamics, Inc.) – This program was developed to predict the effects of cushion acoustic and hull modes on active ride-control systems. The program represents cushion dynamics by a one-dimensional finite-volume model. Effects of lower frequency longitudinal acoustic modes as well as flow lags in fan and vent ducts are included. As in the SES Seakeeping Program, the equations of motions are linearized about selected mean operating conditions. However, the non-linear character of certain phenomena is retained, including side-hull leakage flow and vent-valve position limiting. Active ride-control is simulated as independently-controlled vent valves located at several longitudinal cushion locations. The program has been used to support the detail design of an advanced RCS for the SES-200 and has also been applied to evaluate the potential of "split-cushion" concepts for pitch/heave control and seakeeping improvements for the FRG SES-700 design.
- 6-DOF SES Motion Simulations (Oceanics, Inc.) – The "6-DOF" program developed by Aerojet and Oceanics, and used in development of the RMI 3KSES, is a non-

linear, time-domain simulation of SES motion. The program predicts all rigid-body motions as functions of time from the starting condition. The analytical representations for hull hydrodynamics, bow and stern seals, lift fans, propulsors, and ride-control components are non-linear and considerably more complex than those used in the frequency-domain programs.

- 6-DOF SES Motion Simulations (Textron Marine) – The Textron Marine program is similar to that of Oceanics, except that bow and stern seal dynamics are based on empirical data rather than analytical modeling. Given a realistic starting condition, this program is particularly useful for tracking ship motions during discrete wave encounters and slamming and broaching events. Options have also been developed and incorporated for evaluation of hull structural loads during such events. The evaluation of seakeeping statistics using the 6-DOF programs requires extended run times, plus generation of a time history for a "random" seaway with the desired spectrum characteristics. The 6-DOF program developed by Textron Marine was refined, extended, and extensively applied in support of the 3KSES development effort. Recent applications of this simulation have been relatively limited in scope.
- SES Seakeeping Program (Band, Lavis & Associates) – This design tool was developed by Band, Lavis & Associates (BLA) to be used in conjunction with a whole-ship design synthesis model for SES. The tool was developed from a multiple linear regression analysis of motion data derived from testing a total of 13 SES models of different geometry. This program is currently restricted to predicting heave and pitch motions and accelerations of SES in head seas.
- Time-Domain SES Heave Dynamics Program (Band, Lavis & Associates) – As part of the 1987-1989 SES Hullform Technology Program, BLA developed a time-domain representation of the heave dynamics of air-cushion supported craft. This representation allowed the effects of scale, compressibility, associated water mass and side-hull hydrodynamics to be studied individually. It demonstrated once more that large SES are very much more prone to heave instability than SES of 200 t or less, and that active ride-control will likely be required for these large SES.

A rough comparison of typical large high-speed HSS SES parameters with the corresponding values of typical real SES, such as the SES-200, affecting SES motions and cushion dynamics suggests the following:

1. Non-dimensional cushion parameters for a "typical" HSS design are generally similar to the SES-200 configuration. Levels of wave excitation relative to ship dimensions are, however, significantly smaller. Other significant departures from similarity are as follows:
    a. The wave encounter frequency range of interest is not shifted much relative to the SES-200, while the characteristic frequencies for the overall ship and its systems are substantially reduced. As a result, the characteristic cushion system frequencies (e.g. heave resonance) are close to the wave excitation frequencies and, thus, will be more strongly tuned. The typical cushion instability sources are also moved down closer to the wave excitation range, making filtering and stabilization more difficult. This "bunching" of excitation and resonance frequencies, as

well as the increasing tendency to heave instability, suggests that cushion system design for the HSS SES must be pursued on the basis of an integrated dynamic system including the air cushion, lift system, seals, and ride-control system, as well as the effects of structural dynamics.

b. Increased cushion compressibility at the HSS SES cushion pressures results in a substantially “softer” cushion. This effect impacts the sizing of lift system and RCS equipment, and substantially extends the cushion ”refill” time for on-cushion operation/survival in very large waves.

2. A ride-control system will likely be required for the HSS SES to exhibit acceptable behavior and habitability. Although the relative level of wave excitation is less by a factor of 2, the frequencies of dominant wave excitation (0.2 to 0.3 Hz) and typical HSS SES heave resonance (0.35 Hz) are both within the 0.1 to 0.4 Hz range which result in the greatest incidence of seasickness.

The existing SES computer simulation tools will require some further development to be applicable to large high-L/B SES, but should be capable of providing valuable insights into the effects of scale, which can more easily be accommodated in computer simulation than in the model test tank. The fully-developed programs will be applicable for predicting seakeeping and ride quality of the HSS SES, and for evaluating cushion pressure variations, both statistically and resulting from discrete wave encounters. Issues to be addressed include:

1. Many of the cushion-related resonance and instability frequencies at HSS scale lie close to the wave-encounter excitation frequencies, and cushion damping generally decreases for larger SES. The existing simulation models provide only limited capability to evaluate the effect of these resonances and related control requirements. The available models do not provide adequate coverage to support an integrated approach to air-cushion dynamics and related subsystem design. Development of an improved ability to predict motions and control-system resonances and related subsystem dynamics through systematic development of both frequency-domain and time-domain computer simulations is needed.
2. With the exception of BLA’s Time-Domain SES Heave Dynamics Program, the current programs have neglected as insignificant the effects of cushion-pressure dynamics on the cushion/water interface. The HSS cushion pressures and dwell times are far outside current experience and may no longer be negligible. An expanded development of routines for evaluation of dynamic cushion/water interface effects is needed for incorporation in existing motion prediction programs.
3. A number of issues related to cushion dynamics and control have previously been addressed by Aerojet, Bell, and RMI for the 2K/3KSES, with extensive simulation support. Notable examples include:
   - stern-seal dynamics and flutter
   - waterjet air ingestion in waves
   - lift-fan drive-train dynamics as driven by wave pumping and fan flow control

Development of routines addressing these effects is needed for insertion into motion prediction programs.

To support an integrated approach to SES cushion dynamics, development of "modular" program(s) is needed which will include simulation of basic SES cushion dynamics and motions as a framework for related subsystem dynamics evaluation. Program options will allow emphasis on selected subsystem phenomena and a reasonable level of input complexity when applied to specific dynamic problems such as those noted. The simulations will also be capable of evaluating potential corrections and improvements via the control of cushion dynamics.

3.3.2 Technology Goals

Ship motions of the large slender displacement hulls and high-L/B SES hulls needed for HSS missions will differ from conventional ships. Extensions to the technologies used to predict ship motions for these ships are needed to develop hull designs that will safely and efficiently transport crew and cargo at high speeds in representative sea conditions. These predictions are also needed to design essential subsystems such as SES ride-control systems, lift fans, and bow and stern seals.

A modest extension of displacement monohull seakeeping technology is needed to validate the accuracy of existing seakeeping predictions for hulls with the greater slenderness and larger transoms of HSS hulls.

A more significant extension of seakeeping technology is needed for HSS catamarans and trimarans. Extensions to the monohull-based techniques are needed to reflect the geometric, hydrodynamic, and mass property characteristics of these large slender multihulls.

The SES seakeeping technology development goals are to extend the capabilities of existing motions prediction computer programs to address issues critical to large high-L/B HSS SES designs including cushion resonance/instability predictions, cushion/water interface dynamics, and modular simulations for subsystem dynamics. These technology extensions will allow integrated development of SES hulls, lift systems, seals, and ride-control systems with the seakeeping performance required for HSS missions.

3.3.3 Overview of Development Plan

Displacement hull seakeeping technology development includes extension of monohull tools to address the higher slenderness and large transoms of HSS hulls as well as extensions to model multihulls. The effort consists of software development that incorporates existing model test and full-scale data as well as test data developed as part of this plan.

The scheduling and costing plans for displacement hull seakeeping technology development are shown in Figure 2.2.3-1 for monohulls and Figure 2.4.3-1 for multihulls.

SES seakeeping technology development includes development of cushion resonance/instability predictions, cushion/water interface dynamics predictions, and modular simulations for

subsystem dynamics. Analytical predictions of motions and ride-control performance will be compared with model test data and full-scale experience. The overall program is broken into three phases. The first phase will structure the problem and will develop the basic analytical approaches and routines that can be used to evaluate the feasibility and significance of the expected result. The second phase will incorporate the analysis into one of the existing SES motion simulations. Simulation results will be correlated with available test data. Requirements for experimental validation testing will be established. The third phase will provide experimental validation and demonstration of the simulation predictions.

The scheduling and costing plan for SES seakeeping technology development is shown in Figure 2.5.3-1.

3.4 Maneuvering

HSS displacement hull and SES concepts exhibit several differences from existing ships that affect maneuvering, including operations at high speeds in high seas, use of more slender hulls (higher $L/\nabla^{1/3}$,L/B), use of waterjets with nozzle control for maneuvering, and selective use of small rudders for high-speed maneuvering. These differences are expected to influence initial stability, dynamic stability in waves, and dynamic stability in turns. Factors affecting stability in waves and in turns include the effects of transverse CG shift, roll inertia, forward speed, and turn rate. The objective of the maneuvering and dynamic stability studies is to provide assurance that the large high-speed HSS ships can operate safely throughout their operational envelopes.

3.4.1 State-of-the-Art

The technology to predict the maneuvering and dynamic stability characteristics of displacement hulls is well established. The approach used requires solution of generic equations of motion formulated with empirically or experimentally-derived hydrodynamic coefficients. The resulting system of equations can then be analyzed to assess conformity with U.S. Coast Guard, Code of Federal Regulations, and classification society requirements. While the equations of motion are general, the hydrodynamic coefficients are hullform specific. However, little hydrodynamic data exists for the slender, high-speed, steerable, waterjet-equipped monohulls and multihulls envisioned for high-speed sealift application. Existing displacement hull maneuvering tools are monohull-based and lack the capability to model multihull geometry and mass properties.

Existing regulatory body requirements are heavily biased by the characteristics of slower, less slender monohulls. Structural requirements and crew ride quality considerations often result in these conventional ships reducing speed in higher seas. This combination of current standards and operating practices results in assurance of adequate maneuvering and control authority to assure safe operations for the ship loading conditions, speeds, and sea conditions encountered. However, the greater slenderness, higher speed, large draft variations, and control systems envisioned for HSS displacement hulls may result in development of unstable dynamic behavior modes that do not occur for the more conventionally-designed and operated hulls. Furthermore, the premium attached to speed of HSS ships will encourage maintaining high speeds in high seas. Systematic evaluation of high-speed slender HSS hulls to assess the possible existence of undesirable stability characteristics in calm water and in waves has not been done.

Consequently, while the methodology to analyze maneuvering and control of HSS displacement ships exists, tool extensions to encompass multihulls, additional hydrodynamic data, and analysis are needed to assure safe operations of HSS concepts throughout the operating envelope.

The capability to predict the dynamic stability and maneuvering characteristics of SES via a combination of model testing and computer simulation has reached a high level of maturity. A wealth of test experience has been accumulated over the past forty years, but, until recently, this has mostly been limited to the characterization of specific designs with little attempt or opportunity to explore, systematically, any wide variation in hullform or basic stability parameters. Even the model testing which followed the only known capsize of an SES (the U.S. Navy's experimental test craft, XR-1, on the Delaware River in December of 1964) was limited principally to the exploration of craft beam and side-hull deadrise. The beam of the XR-1 was increased to improve roll stability as a result of the model tests.

Many of the early analyses of the dynamic stability of SES addressed stability using linearized equations of motion. The studies were limited to calm-water operation and considered either longitudinal or lateral stability modes. In general, the essential properties, or possible modes, of transient response and stability were determined by the nature of the roots of the characteristic equation. Although, in most cases, forces and moments were decidedly non-linear, dynamic stability could at least be assessed for small angular displacements. These early studies were concerned, therefore, not so much in predicting the ultimate non-linear response, but rather with predicting those conditions and configurations for which unstable behavior could build up so that such motions (and configurations) could be avoided.

In recent years, advances in computer-aided analysis have permitted more extensive procedures to be developed for treating the non-linear behavior of the SES. With such tools and testing techniques, SES can be designed to exhibit adequate static and dynamic stability in both the intact and damage condition while both cushionborne and hullborne. When hullborne, this is due to the large initial waterplane moment of inertia provided by the wide separation of the side-hulls and the relatively small clearance of the wet-deck, which results in the cross-structure entering the water after only a few degrees of list. The resulting increased waterplane limits the impact of off-center flooding and sinkage; consequently, larger subdivision lengths are acceptable on SES designs than on equivalent-sized monohulls.

As SES become larger, for ships of moderate speed, the preferred length-to-beam ratio tends to increase on account of the advantages gained in the form of reduced resistance. High cushion heights are also desirable for large ocean-going SES to keep the wet-deck clear of large waves. High wet-deck heights tend to imply high vertical CGs. The combined effect has been to develop high, narrow ships for which on-cushion dynamic roll stability during turns and in synchronous beam seas, especially in adverse weather, has become of greater concern.

In recognition of this trend, recent large SES designs have generally featured side-hulls of relatively larger volume to increase stability and, in addition, they have been able to accommodate heavy machinery relatively low within these side-hulls to lower the center of gravity. For the range of small SES built to date, the side-hulls have generally been too small for the

installation of much machinery, which must instead be located above the level of the cross-structure wet-deck, which has resulted in a relatively higher vertical center of gravity (VCG). In addition, for large SES, all the fuel is located in the lower extremities of the side-hulls to help lower the VCG in the full fuel-load condition.

Important features affecting dynamic stability also include the side-hull length, volume and deadrise, the types of bow and stern seals, the size and location of skegs, fences and rudders (if included), the type of propulsion system, the type of maneuvering system, and the ship's moments of inertia.

The primary circumstances leading to a risk of unfavorable dynamic behavior include high-speed turning maneuvers, sudden helm reversal and/or sudden propulsor or steering-system failures at high speed, running with high winds and synchronous seas on the beam, and operation in very steep following or quartering seas.

Research conducted in the UK has generated a greatly improved understanding of overall on-cushion stability requirements to the extent that provisional criteria based on practical and purely numerical methods have been set. The discussion which follows is based on the results of this research and, therefore, represents the most up-to-date analysis currently available.

The research using capsizeable radio-controlled models and towing-tank models has shown that the principal challenges to be addressed in assessing the on-cushion stability of an SES are the behavior in high-speed turns when the vessel is subjected to a large overturning moment and the behavior in beam wind and sea conditions when resonant rolling can cause capsize.

Model tests have been correlated with full-scale trials and, from both techniques, information about hazardous situations that might arise and how to counteract them has been obtained for the practical benefit of operators and commanders. The overall conclusions regarding stability in waves are:

- Capsize in a seaway is associated with a resonant type response of ship to wave. The capsize sequence is similar regardless of the type of wave system encountered. Radio-controlled model tests are a valuable means of determining the most critical situations. However, they have proved to be an inefficient and time-consuming method of assessing the critical VCG compared to static beam-sea tests in the towing tank. Current indications are that the absence of forward speed in such tests does not measurably affect the results.
- Capsize of an SES in a seaway is most likely to occur when beam-on to wind and sea, and may be preceded by an obviously resonant build-up of roll angle. It may also happen without warning. Each SES design has a particular roll inertia with a critical VCG, below which no capsizes have been observed in realistic operating conditions. A model test method has been evolved which enables the effect of VCG on the safety of a design to be established quickly. Using this technique, the effects of major design parameters on the critical VCG can be evaluated.

- It is suggested that ships should not be operated with a ratio of VCG-to-beam above a value of about 0.25 to 0.30. All of the model capsizes involved VCG ratios above these values by a significant margin, which is required to allow for the effects of beam winds and breaking wave crests and the expectation that steeper waves may be encountered at sea. It is to be noted that all full-scale ships considered have VCG ratios below these values and have operated extensively and safely.

The overall conclusions regarding stability in turns are as follows:

A significant reduction in roll stability occurs in high-speed turns. Negative roll stiffness may be generated over a range of roll attitudes near the upright, even though the vessel has positive stability on a straight course. Such negative zones appear to be the cause of large amplitude coupled roll/yaw oscillations that have been observed at both model and full-scale and have been simulated mathematically. Whether a zone of negative roll stiffness in turns exists may not become apparent until a sufficient external roll moment causes a sudden, large change in roll attitude. Roll stability in turns is adversely affected by reductions in bow-up trim and by increases in lift air flow or rate of turn. For most SES, sudden helm reversals will not produce critical behavior.

Considerable progress has been made towards a comprehensive understanding of the dynamic stability of SES. However, areas have been identified that would benefit from further examination during the design-development phase. Some of these are discussed below.

Initial Stability

Cushion air flow has been shown to have a substantial effect on initial roll stiffness, both with and without forward speed. Further work is needed to develop air flow leakage models, especially over the first 10 degrees of heel, to improve maneuvering and control simulations.

Stability in Waves

Testing in beam seas has not been conducted with a transverse center of gravity (TCG) shift of greater than 2% of beam. This figure was derived from an examination of levels of TCG that are likely to occur in normal operational circumstances. However, in extreme conditions, when large roll angles are being experienced, major movement of payload may occur which could result in TCG shifts of up to 6%. This would significantly reduce the critical VCGs already determined.

Roll radius of gyration has a significant and non-linear effect on critical VCG. Further information on how the critical VCG of the high-L/B hullforms of interest is affected by roll inertia is needed.

Although open sea tests have indicated that forward speed has little influence on the conditions required to cause capsize, the comparisons that have been made are not at all precise. Since craft hydrodynamics at speed are different than those when stationary, a more quantitative comparison in controlled conditions is required. In conducting such experiments, the potential effect of increased frequency of encounter due to a slightly diagonal, up-sea heading relative to waves should also be explored.

Stability in Turns

Because of its fundamental effect on the possibility of negative roll stiffness, a better understanding of why and how roll stability changes from zero speed to maximum speed is required to improve numerical methods of assessment.

Since stability in turns is adversely affected by increased rate of turn, better methods of predicting turning-circle diameter are needed. This, in turn, is closely related to longitudinal trim and directional stability. Examination of the effect of a wider range of craft trim angles is needed, particularly because there are indications from full-scale trials that excess bow-up trim may also be disadvantageous.

Analysis

Because of the multiplicity of relevant parameters, it is not practical to model test all possible combinations. However, a sufficient understanding of capsize behavior has been obtained to permit the development of a mathematical simulation of an SES rolling in regular beam seas. Once proven against the widespread body of data now available, such a simulation would be an invaluable tool, both for assessing individual designs and for investigating the effects of wave slope, roll inertia, and increasing TCG.

While a method of analysis has already been derived, more sophisticated techniques for calculating the various components of roll moment in turns are needed. In particular, a means of evaluating the way in which hull shape affects the variation of planing forces with trim and roll could improve the accuracy attainable.

Validation of existing methods, and methods under development against existing designs, is needed.

3.4.2 Technology Goals

Maneuvering and control characteristics of the large slender displacement hulls and high-L/B SES hulls needed for HSS missions will differ from existing ships. Extensions to the technologies used to predict maneuvering and control performance for these ships are needed to develop hull designs that will safely and efficiently transport crew and cargo at high speeds in representative sea conditions. These predictions are also needed to design essential subsystems such as steerable waterjets and high-speed steering rudders.

Modification of existing displacement hull maneuvering analysis tools is required to model multihull geometry and mass properties and include the effects of large steerable waterjets. Modest extension of maneuvering technology is needed for monohull and multihull displacement hulls to generate the hydrodynamic data needed to make predictions for the slender, high-speed HSS hulls. Analysis and testing of HSS hulls are required to identify and eliminate undesirable modes of transient response in calm water and in high seas.

SES maneuvering technology development goals are to provide the capability to develop SES for the HSS mission with adequate maneuverability and controllability in all modes of operation

(on-cushion, off-cushion and partial-cushion) and applicable sea states. Issues to be resolved include the amount of required steering, reversing capability, and performance in quartering seas.

3.4.3 Overview of Development Plan

Displacement hull maneuvering technology development includes extension of monohull tools to model hull geometry and mass properties of multihulls as well as generation of hydrodynamic data for the slender, high-speed HSS hulls. The effort consists of software development that incorporates existing model test and full-scale data as well as test data developed as part of this plan.

The scheduling and costing plans for displacement hull maneuvering technology development are shown in Figure 2.2.3-1 for monohulls and Figure 2.4.3-1 for multihulls.

SES maneuvering technology development consists of extending existing SES capabilities through a program of design, analysis and model testing. Major issues to be resolved are the effects of cushion airflow on initial stability, the effects of transverse CG shift, roll inertia, and forward speed on stability in waves, and the effects of forward speed and rate of turn on stability in turns. Data from existing craft, including the SES-200, TSL-A and others, will be used to validate analytical simulations at larger than model size.

The scheduling and costing plan for SES Maneuvering is shown in Figure 2.5.3-1.

4.0 LOADS, MATERIALS, AND HIGH-STRENGTH/LIGHTWEIGHT STRUCTURES

4.1 Introduction

To design a marine vehicle, one must know the seaway and cargo loads, as well as the structural response and the properties of the structural materials. Although a great deal is known about the seaway loads on conventional ships (Figure 4.1-1) operating at slow to moderate speeds, the effects of high speeds are unknown. Similarly, the seaway loads on novel hullforms are currently unknown. To attempt to design a ship outside of our current experience base, that is, operating at very high speeds or with a novel hullform, without model tests can lead to two unacceptable states: (1) under-predicting the loads so that the ship suffers significant (and possibly catastrophic) structural failure, and (2) applying excessive factors of safety (to cover ignorance levels) leading to an overly heavy structure.

Figure 4.1-1: Seaway Loads

The materials and structure required to resist the seaway loads have historically been dictated by cost, weight, and producibility considerations. Moderate to large size conventional ships are built of steel, a relatively low-cost but high-weight material. Smaller, weight-critical craft have been built out of aluminum or fiber-reinforced plastic (FRP) composites since the 1950s. Until now, however, there has not been a need to go to the lighter and more costly materials for larger ships. The lightweight materials can save weight, which can be used to increase the ship's speed, range or payload. Since the structural weight of a ship is a large part of its displacement, the potential payoffs in weight savings are substantial – in the thousands of tons for large ships. To realize these weight savings, a significant research and development effort is necessary to resolve a number of issues. Projected weight savings and corresponding deadweight density increases are shown in Figure 4.1-2 for near and far-term high-speed sealift ships and compared with that of existing ships.

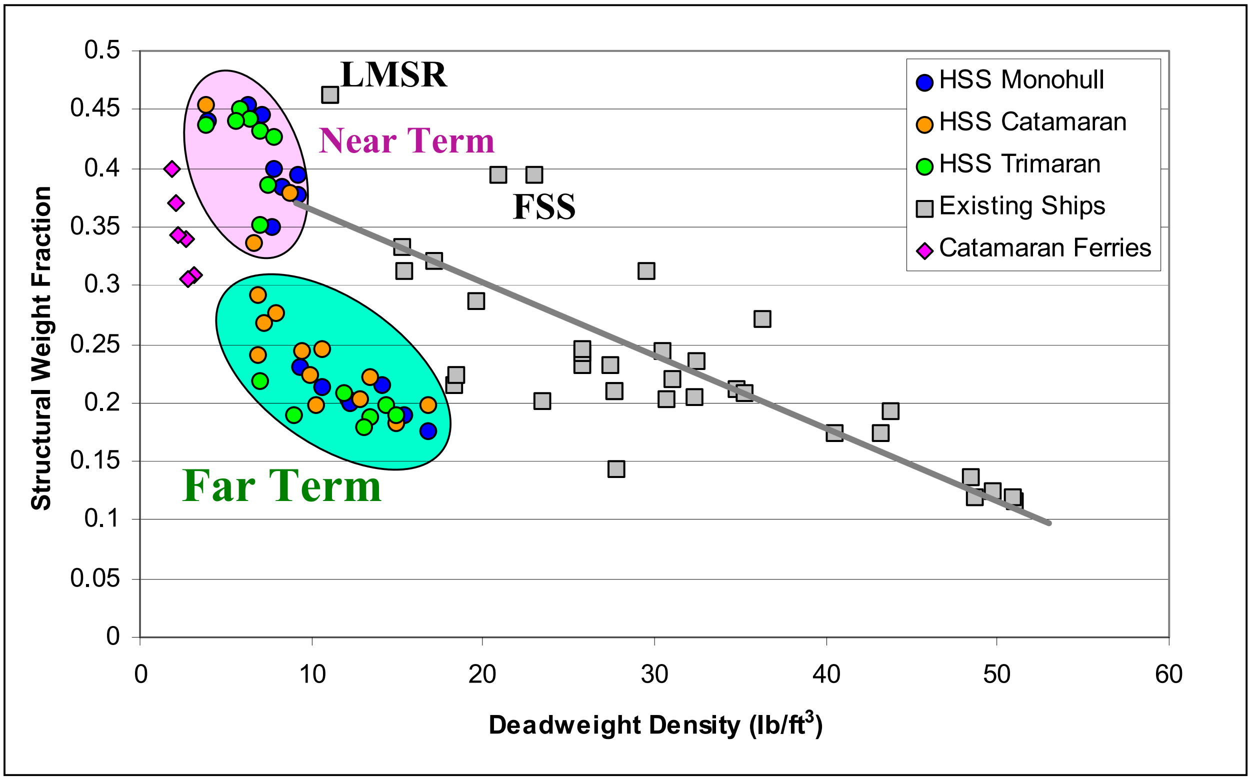


Figure 4.1-2: Structural Weight Fraction versus Deadweight Density

Numerous research and development issues related to loads, materials, and high-strength/ lightweight structures need to be addressed to realize these weight savings. These issues are outlined in Figure 4.1-3 with approximate effort levels identified. The individual tasks will be described in greater detail later in this section.

Time to complete (years)	1	2	3	4	5	6	7	8	9	10	Est. Cost ($K)
Identify Structural Concepts and Materials											3,600
HSS Guide/Navy/Commercial Criteria											1,250
- Loads testing *(cost included with hull form technology)*											0
- Conditioned-based monitoring and inspection											8,000
Near Term Structures											
- Develop and design integrated decks and ramps											11,000
- Fabricate and test large-scale demonstration											8,000
- Fabricate, install and test at sea											12,500
Far Term Structures											
- Develop optimum materials and joining technology											18,000
- Develop full-scale design											1,400
- Design, build and join full-scaled modules											185,800
- Module testing and evaluation											6,500
Funding ($K)	1,050	9,750	14,850	53,250	78,550	70,950	20,850	4,850	1,950		256,050

Figure 4.1-3: High-Strength/Lightweight Structures Technology Development Plan

4.2 Seaway Loads

There are two kinds of seaway loads acting on ships: primary and secondary. Primary loads are bending and torsional moments which flex and twist the hull as if it were a beam or girder. The interaction of the wave buoyancy forces and the weight of the ship cause bending in the vertical plane (hogging and sagging); see Figure 4.2-1. Bending in the transverse plane (lateral bending) and torsional twisting is caused by port to starboard differential buoyancy and rolling in oblique seas. Transverse plane loads are particularly important for multihulls and SES.

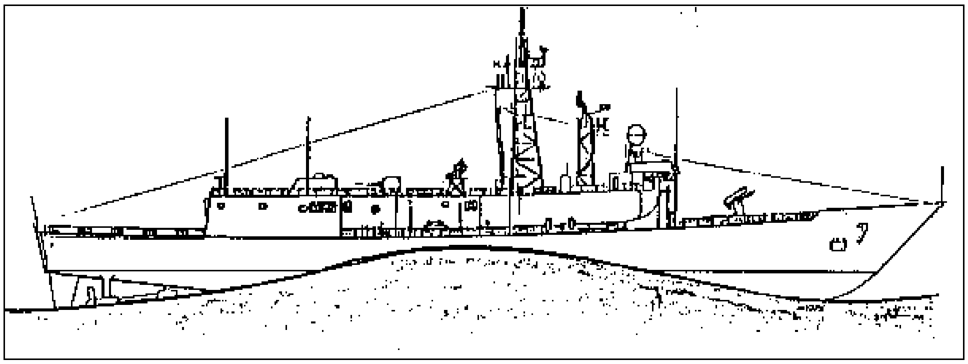


Figure 4.2-1: Hogging Bending Moment

Historically, conventional monohulls have been designed by developing shear and bending moment diagrams from a static balance of the hull girder on a standard wave. Damage during sea trials of the CVA 9 Essex in the late 1950s led to a series of full-scale trials and model tests to define a dynamic component (whipping) from slam impacts that increase the vertical and lateral bending moments along the length of the ship. The slam-induced whipping is exacerbated by speed and, in some cases, can approach the magnitude of the wave-induced moments. All of the primary hull girder moments increase in proportion to the square of the length of the ship.

Secondary loads are the static and dynamic pressures acting on local structure. Hydrostatic pressures are caused by the head of water from passing waves and are functions of ship draft and sea state. At slam impacts, a hydrodynamic pressure is caused by large bow motions (pressures then act on the bottom of the bow as it re-enters the sea, when the bow flare is immersed, or when multihull/SES cross-structures are immersed). Wave slapping pressures can be significant on the hull sides and transom. Green sea loadings occur when waves crash over the bow, striking the weather deck and front of the deckhouse (see Figure 4.2-2). All of these hydrodynamic pressures are functions of hull geometry and increase with ship speed and sea state.

Figure 4.2-2: Green Sea Loading

4.2.1 State-of-the-Art

There are a number of analytical tools available for predicting seaway loads for conventional monohulls. SMP95 is a linear strip theory code in the frequency domain that gives good results for the wave-induced portion of hull girder bending, but is not applicable for whipping effects. QLSLAM, DYNRES, and LAMP are time-domain codes that have the potential for including whipping in hull girder bending, but all are limited in one way or another. SLAM-2D can predict bow slamming pressures. All of the analytical codes were developed for conventional monohulls and have limited validation. Extension of this conventional monohull technology is needed to address the greater slenderness and higher speeds of HSS monohulls. Additional extensions to the technology are required to model geometry and mass properties of HSS trimarans and catamarans. Model test data is required for HSS displacement hull concepts to guide development of the analytic models and validate the predictions. They need further validation (and possibly modification) for applications to novel hullforms.

Model tests can be used to predict primary and secondary loads for conventional and novel hullforms under extreme sea and operational conditions. The test data are analyzed and presented in a probabilistic format that can account for such variables as expected lifetime, sea conditions, and operational parameters.

4.2.1.1 SES Issues

High-speed sealift SES will represent a large increase in size from previous SES hulls, the largest of which is, at present, 1,500 tons. As with displacement hullforms, the high design speed will increase the magnitude and frequency of slam loads compared to a conventional ship. In addition, the traditional spread between hull primary response frequency and the wave encounter frequency will narrow. These factors may result in slamming becoming a major contributor to the determination of design primary bending moment.

While slamming will contribute to primary design bending moments, it will contribute in a different way than it does for more traditional hullforms. The bow seal and the cushion protect the SES hull from slams at speed. However, operation without the bow seal or the cushion at low speed will still result in slam loads that must be considered in the primary design bending moment.

Although full-scale SES experience is limited to hulls of considerably smaller size than those required for HSS concepts, a large body of experimental, analytical and design data has been developed for much larger SES concepts. In the U.S., comprehensive development of the SES concept began with the SES-100A and SES-100B programs in 1965. As these programs evolved, a number of different approaches were explored to develop rational design loads for SES. Initially, because there was no historical experience, a linear, frequency-domain seakeeping model was used to predict the frequency of bow and wet-deck slams and the associated relative velocity at impact. This information was used as input to a 6 degree of freedom, time-domain impact model that used an adaptation of seaplane theory to predict pressure distributions, total loads, motions and accelerations. Limited full-scale trials of the SES-100A and SES-100B were used to validate predicted values. Later, during the 2KSES program, bending moments and stresses were measured during segmented and “grillage” model tests, and a full-scale section of the wet-deck ramp was tested to determine the stresses that would be experienced during high-speed impacts. The “grillage” model was built to model the structural elastic characteristics of the full-scale RMI 2KSES and was tested in the on-cushion and off-cushion models of operation at all headings to the waves. These tests showed that few wet-deck impacts were experienced in the on-cushion condition. However, the loads and bending moments experienced from wet-deck impacts in the low-speed off-cushion condition were found to be considerably higher than the high-speed on-cushion loads.

4.2.2 Technology Goals

As hullforms are introduced outside of our current experience base, model tests will be required to determine the proper loads and response. These model tests will expand the experience base, enhance our analytical capabilities, and lead to a reliable, efficient hull structure.

Although model tests are the key to understand the loads and design an optimum structure to resist the loads, a number of load reduction strategies to reduce the primary and secondary seaway loads are also being considered to reduce the hull structural weight. For example, hull or bow forms that reduce slamming and/or primary loads can save structural weight as well as improve seakeeping and resistance.

Condition-based monitoring methods are also being utilized and developed to reduce the structural weight. Implementing wave measurement systems to avoid incoming waves can reduce primary and secondary loads. Active systems are being investigated to reduce the whipping component of the hull girder bending moment. In addition, the design allowable stresses can be relaxed if frequent, focused inspection schedules are conducted, automated hull inspection and repair systems and techniques are implemented, and strain gauges and sensors are used to monitor the hull structural behavior during its operation. Further research and

development in these areas is needed to realize the potential weight savings and the impact on the structural reliability.

4.2.3 Development Plan for Loads

The structural loads and response of large ships and novel hullforms at very high speeds are unknown and need to be determined to avoid over-design or catastrophic failures. Because of the need for a very low structural weight fraction, optimal structural performance is required for high-speed sealift missions, making the determination of these unknown loads very important.

The increased understanding of the structural loads and response leads to the development of design guidelines, the investigation and implementation of load reduction strategies, and the development and adoption of active, strain monitoring systems and focused inspection schedules to provide a reliable and optimum lightweight structure.

The effort needed to develop design guidelines, determine structural loads and response, and develop load reduction strategies and monitoring procedures and systems to ensure a reliable, lightweight structure is shown in Figure 4.1-3. This plan includes:

- Model tests of various hull stiffnesses and geometries, speeds and headings to determine primary and secondary loads for high-speed operations and novel hullforms.
- Analytical codes verification and modification.
- Investigation and development of active systems to reduce the whipping component of the hull girder bending moments.
- Development of wave measurement systems and load avoidance and monitoring strategies that will reduce hull girder loads and, therefore, structural weight.
- Investigation of structural modifications such as an articulated hullform to reduce the primary loads.
- Development of automated hull inspection and repair systems, implementation of focused inspection schedules, and the analysis of their effect on structural weight and reliability.
- Development of design guidelines, using the results of the various investigations, optimizing structural reliability and minimizing weight.

4.3 <u>Materials</u>

4.3.1 State-of-the-Art

Large conventional monohull ships are predominantly constructed of steel, while smaller weight-critical vessels (under 130 meters) are frequently constructed of aluminum or composites. Similarly, steel is the material of choice for many larger SES concepts such as the German SES-700 and the U.S. Navy's Intra-Theater Sealift Ship design. However, most of the smaller SES constructed to date have used aluminum as their basic structural material. Exceptions include the

Vosper Hovermarine SES (the HM-2 and HM-5 series) that are constructed principally of glass-reinforced plastic and the Chinese 95-ton 719 which is primarily composed of steel. Weight-critical ships frequently use aluminum material to reduce weight because it has one-third the density and modulus of steel and a fatigue allowable stress one-half that of steel. At a first level approximation for ships governed by hull girder bending (ships over 130 meters long), aluminum can save one-third of the structural weight of a steel vessel. There are no technical reasons why large ships cannot be fabricated from aluminum, but consideration must be given to the relatively low fatigue characteristics of aluminum and the large deflections that aluminum structures exhibit compared to steel structures. However, the cost of aluminum is five to eight times more expensive than that of steel, and aluminum has a relatively low resistance to fire. For a non-combatant, fire protection would be required in a few key locations.

Many ships are constructed of high-strength steels or use high-strength steels in certain locations. Some of the high-strength steels are twice as strong (yield strength) as ordinary steel, yet they do not save much weight in large ships. The reason is that structural details composed of high-strength steels have almost the same fatigue allowable stresses as ordinary steel, and, hence, these ships require just as much material to resist hull girder bending. The extra strength can only be used to resist secondary loads. The improvement in the fatigue characteristics of high-strength steels are necessary to significantly improve the structural weight fraction.

4.3.1.1 Composites

Composite structures consist of fiber reinforcements (such as E-glass or carbon) encapsulated in a resin matrix (such as vinyl ester or phenolic). Composite materials can be used to produce single-skin, stiffened, or sandwich structures; see Figure 4.3.1-1. They have been used for primary structures on small craft or vessels for many years. They are also applicable for secondary structures such as decks, foundations, doors, hatch covers, enclosures, deckhouses, stacks, and masts.

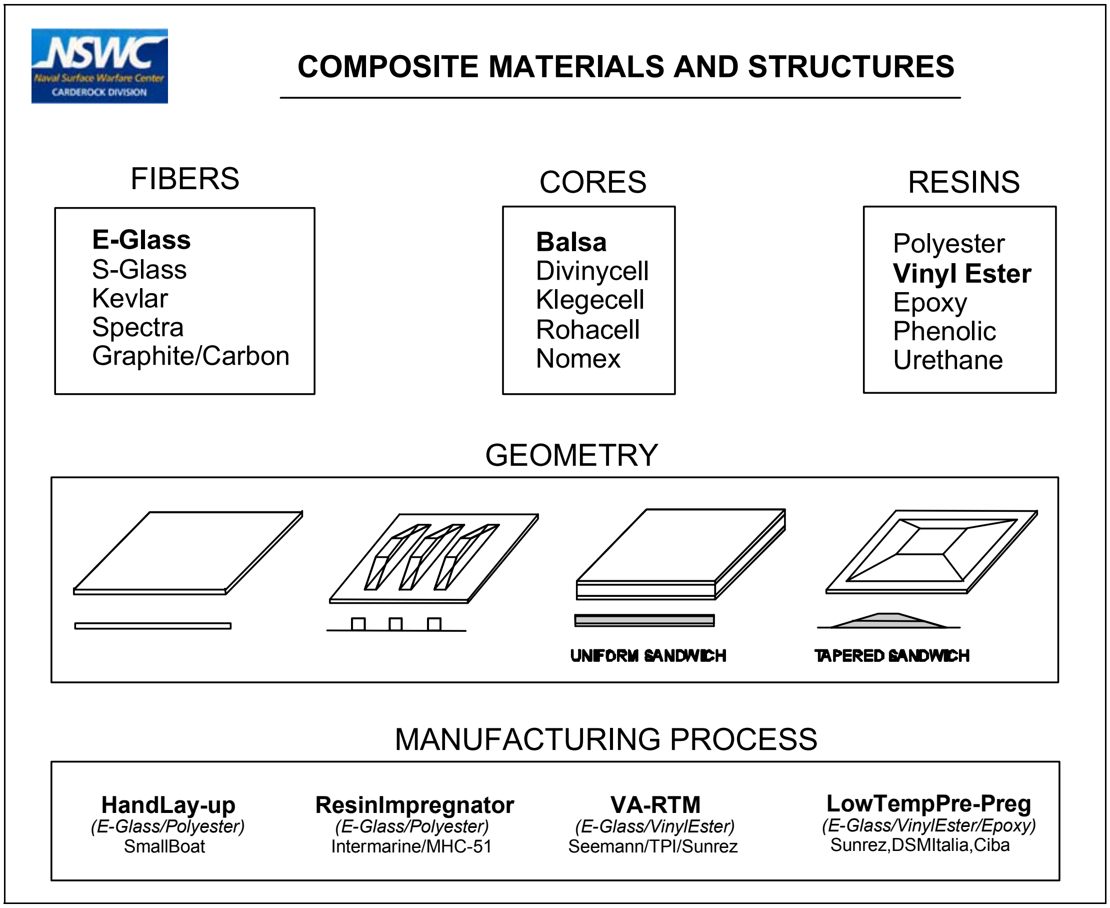


Figure 4.3.1-1: Composite Materials and Processes

4.3.1.2 Titanium

Titanium has a yield strength higher than most high-strength steels, with a density of only 57 percent that of steel. The potential weight savings exceed that of aluminum and it has a much better fire resistance. Titanium alloys have been used extensively in the aerospace industry in the United States and have received some attention in the automotive industry. Timetal 10-2-3 (Ti-10V-2Fe-3Al) is used in the main landing gear of the Boeing 777. Timetal 15-3 (Ti-15V-3Cr-3Sn-3Al) has been used in environmental control system ducting, firefighting bottles, door springs, and small nut clips. This alloy has good formability and was used for more than one hundred formed parts on the B1B bomber. The superplastic alloy SP-700 has been used in place of stainless steel in steam turbine blades, hand tools, and golf club heads.

Titanium has already been used for submarine hulls in the former Soviet Union. The biggest issue with titanium is its cost and availability. Titanium is thirty times more expensive than steel and it must be imported from Russia. In addition, it is more difficult to weld.

4.3.1.3 Exotic Metals

Magnesium and beryllium alloys are two of the more exotic metals, which have not been used on ships, but have far-term potential:

- Magnesium-based alloys exhibit very low densities and high specific strengths. The density is approximately two-thirds that of aluminum and one-fifth that of steel. It has the best strength-to-weight ratio of any cast metal and exhibits good damping capacity, castability, machinability, corrosion resistance, and electromagnetic interference shielding properties. The chemical compositions and tensile properties of the most common Mg-based casting alloys are presented in Table 4.3.1-1. Mg-Al-Mn alloys constitute nearly 90 percent of all structural applications of magnesium. However, they are unsuitable for use above 150° C due to poor creep strength. Mg-Al-Si-based alloys, such as AS41A, exhibit improved creep resistance up to 175° C while still maintaining good elongation, yield strength, and ultimate tensile strength. Although magnesium is subject to galvanic corrosion, the susceptibility can be reduced by careful control of alloy chemistry.

- Beryllium-aluminum alloys containing greater than 60 percent (by weight) beryllium are favorable materials for applications that require light weight and high stiffness. These alloys have 22 percent lower density than aluminum, with three times the elastic modulus and a 40 percent lower coefficient of thermal expansion. Some aluminum-beryllium alloys have been developed for aerospace applications. The AlBeMet series of alloys was initially selected for a folding fin on the SR-71 Blackbird. The properties of some of the beryllium-aluminum alloys are presented in Table 4.3.1-2. AlBemet 162 is a promising alloy with a high-cycle fatigue limit of 10^7 cycles at 30.5 ksi. It is immune to stress-corrosion cracking at 90 percent of yield stress in saltwater at 65° C for 169 hours and in salt at 315° C for 100 hours.

Table 4.3.1-1: Properties of Cast Magnesium Alloys

Alloy	Composition (% Weight)			Tensile Properties		
	Al	Mn	Zn	Yield Strength (MPa)	Ult. Strength (MPa)	ε (%)
AZ91D	9.0	0.13	0.7	150	230	3
AM60B	6.0	0.13	-	115	205	6
AS41A	4.3	0.35	-	150	220	4
AE42	4.0	-	-	110	244	17
AZ91E-T6	8.7	0.13	0.7	145	275	6
SE41A-T5	-	-	4.2	104	205	3.5
ZC63-T6	-	0.25	6.0	125	210	4.0

Table 4.3.1-2: Properties of Beryllium and Aluminum Alloys

Alloy	Elastic Modulus (GPa)	Density (g/cc)	Thermal Cond. (W/m-K)	CTE (ppm/K)
AlBeMet 162	200	2.1	210	13.9
AA6061	69	2.8	170	23.6
Beryllium	300	1.8	210	11.5
Alum-Lithium	90	2.5	120	23.6

4.3.2 Technology Goals

In the struggle to develop a low-cost, high-strength/lightweight material, several obstacles remain. Stiffness and fire performance are issues that must be addressed. Effective repair procedures that ensure structural integrity must also be developed. Material development costs can be significant and consideration must be given to production-mode acquisition costs. Many of the materials require strict environmental control during fabrication, requiring significant capital investments in infrastructure development.

Conventional composite fabrication processes are critical in quality control. Because of this issue, new fabrication processes (see Figure 4.3.2-1) such as vacuum-assisted resin transfer methods (VARTM) have been developed to provide more consistent quality control from part to part. However, variability in material properties continues to be an issue and is highly dependent on the manufacturing process selected. Worker skill also continues to play a significant role in the quality and consistency of the resulting composite material. Further research and development is needed to develop low-temperature, low-cost/high-quality manufacturing processes and fiber/resin combinations that minimize material property variation and maximize strength and stiffness characteristics.

Although stiffness is not as critical for secondary structures or for primary structures when the ship length is less than 130 meters, when the hull length starts to exceed 130 meters, stiffness becomes more of a concern for virtually all of the non-steel material options currently under consideration. For the primary hull structure of large ships, the limited stiffness of non-steel materials can yield a large hull deflection, which may be problematic for critical alignments. Maintaining hull girder stiffness may be required to avoid hull resonance issues such as springing and whipping.

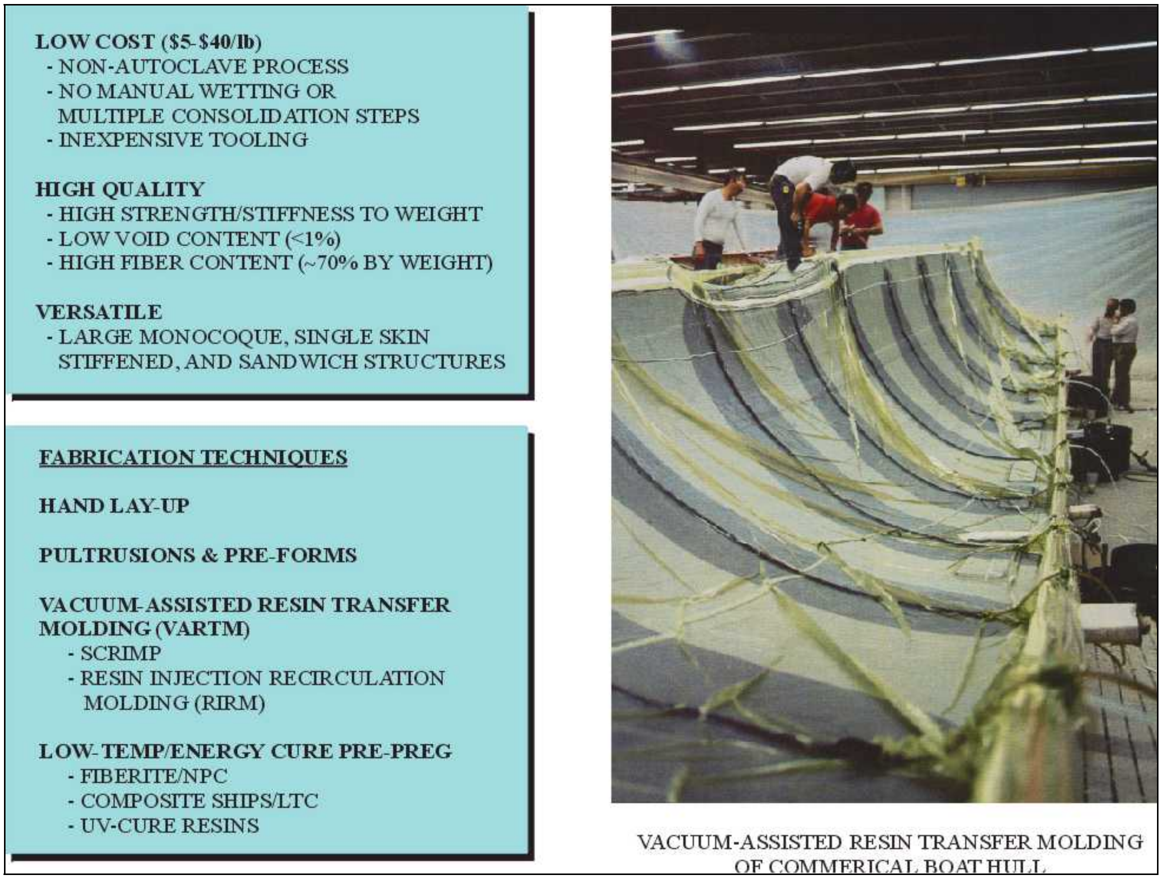


Figure 4.3.2-1: Composite Fabrication Techniques

In the near-term, E-glass and carbon composites are effective in reducing weight in secondary structures, but they have a low stiffness for the primary hull structure bending in large ships (over 130 meters). In the long-term, carbon fiber improvements or more exotic fibers will provide increased stiffness to composite materials. Furthermore, advanced hybrids of composite and metallic materials may be applicable for primary structure of large or very large ships (over 300 meters).

Fatigue characteristics must be improved with many of the material options. The fatigue limitations of aluminum and high-strength steels result from their as-welded properties. Improved welding methods (or eliminating welding by adhesive joining methods) can increase the fatigue allowable stresses for both aluminum and high-strength steels. For example, flush ground welding of aluminum increases the fatigue strength to two-thirds that of ordinary steel, resulting in a fifty-percent structural weight saving. Weight savings for high-strength steels would be proportional to any increases in fatigue allowable stresses from advanced welding/ joining techniques. Such advanced welding and joining techniques need to be investigated and developed, and are certainly possible in the far-term.

The current fatigue database of titanium components is inadequate to ensure a reliable titanium ship design. Fatigue tests of welded titanium components are needed to develop design criteria

in order to design a titanium ship in the mid to far-term timeframes. Since titanium is non-magnetic, new non-destructive inspection (NDI) methods must be developed to replace the common magnetics-based inspection methods currently in use for steel.

4.3.2.1 Summary of Material Properties

The following is a summary of the material properties as they currently exist. Relative stiffness in Table 4.3.2-1 is represented by Young's Modulus (Modulus of Elasticity).

It is expected that, as composites become more widespread, their unit costs will decrease in the far-term. The following unit costs are raw material costs only; producibility issues and fabrication costs are not included in this study. In general, the structural material costs of a steel ship are very small compared with fabrication, installation, and equipment/machinery costs. The average costs for cutting, welding, rigging, painting, and material is on the order of \$25 to \$30 per pound for a steel combatant and half that for a commercial ship. Thus, in the future, the total fabrication costs would likely be much closer for all of these items than are the material-only costs of Table 4.3.2-1.

Table 4.3.2-1: Material Properties Summary

Material	Density Lb/ft^3	Yield Strength Ksi	Young's Modulus Ksi	Fatigue Stress Ksi	Fire Resistance	1995 Costs \$/Lb
ABS Grade A steel	491	34	29,600	20	Good	0.29
ABS Grade AH steel	491	55	29,600	20	Good	0.34
Aluminum (5086-H34)	166	16-22	10,000	10	Poor	1.65
Titanium	280	140	16,500		Fair	10.00
Sandwich Panel-LASCOR (stainless steel)	245-320	55	29,600	20	Good	
Composite Resins						
- Vinyl Ester	70	11-12	490		Poor	1.74
- Phenolic	72	5	530		Good	1.10
- Epoxy	75	7-11	530			3.90
Composite Fibers						
- E-glass	162	500	10,500			1.14
- S-glass	155	665	12,600			5.00
- Carbon-PAN	110	350-700	33-57,000			12.00
- Kevlar 49	90	525	18,000			20.00
Composite Cores						
- Balsa	7	1.3	370	N/A	Insulator	3.70
- Honeycomb Nx HRH-78	6	N/A	60	N/A		13.25

Material	Density Lb/ft^3	Yield Strength Ksi	Young's Modulus Ksi	Fatigue Stress Ksi	Fire Resistance	1995 Costs $/Lb
Composite Laminates - Solid Glass/Polyester - Solid Glass/Vinylester - Solid Carbon/Epoxy	96 90 97	20 50 88	1,400 3,000 8,700	Excellent fatigue life	Poor Fire Resistance – Good Insulator	2.50 3.50 10.00
Composite Sandwich - Glass/Poly Balsa Sandw. - Glass/VinE PVC Sandw. - Carbon/Epoxy Nomex	24 18 9			Excellent fatigue life	Poor Fire Resistance – Good Insulator	4.00 5.00 20.00

4.3.3 Development Plan for Materials

The high-speed sealift missions require the further development of low-cost, high-strength/ lightweight materials. For secondary structures and for the primary structure of ships less than 130 meters, several lightweight/relatively low-stiffness materials are already being used in particularly weight-critical applications. Cost considerations often dictate the use of steel construction for components that are not weight-critical.

The material of choice for the primary structure, as ship lengths exceed 130 meters, remains steel. The primary reasons for the selection of steel include cost, stiffness, fatigue performance, fire performance, property variability of certain other material options, and shipyard experience with steel.

Initially, all of the material options being considered for high-speed sealift applications must be investigated. Eventually, many of the options will be removed from further consideration because of insurmountable issues that are discovered during the investigations. The "weeding out" process is necessary and unpredictable, and will reduce the number of material options available for certain applications.

Although it is often difficult to separate material developmental issues from structural developmental issues, a summary of the research and development effort was shown earlier in Figure 4.1-3 and includes:

- The development of improved welding and joining technology for improved ultimate strength and fatigue performance (all materials).

- The development of cost-effective, structurally-sound repair procedures (all materials).
- Research to improve material stiffness characteristics (all materials).
- Detailed cost-benefit analysis identifying acquisition and life-cycle trends.
- The development of low-cost methods to meet fire containment and toxicity criteria (all materials).
- The further development of a reliable, low-temperature, curing process (composite materials).
- The development of a manufacturing process and material composition that yields high strength while ensuring consistent material properties (composite materials).
- The development of high-strength/stiffness fibers and resins (composite materials).

4.4 Structural Concepts

4.4.1 State-of-the-Art

Ordinary steel and high-strength steel stiffened panels continue to be the standard for large ship primary and secondary structures. LASCOR and composites are increasingly being used in secondary structures to reduce weight when necessary. These technologies are currently under investigation for use in the primary structures of large vessels.

For ships less than 130 meters in length, aluminum plate-stiffener construction and composite construction are the choices for the primary and secondary structures when minimum weight must be achieved. Fatigue strength of the aluminum vessels has been an issue, and careful monitoring is of importance.

4.4.1.1 Sandwich Metals (LASCOR – laser-welded corrugated core)

Sandwich metal structures consist of two thin face sheets of metal joined together by a corrugated core; see Figure 4.4.1-1. The separation of the face sheets provides high bending stiffness at a low weight. Stainless steel LASCOR panels have been used on Navy ships for over a decade to save weight for platforms, hanger doors, and deckhouse enclosures; see Figure 4.4.1-2 for past and proposed LASCOR applications.

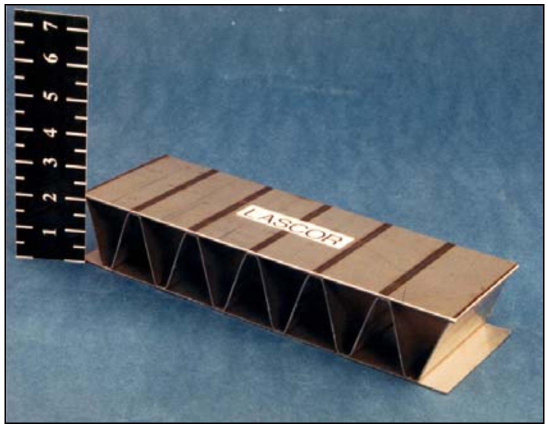


Figure 4.4.1-1: LASCOR Technology

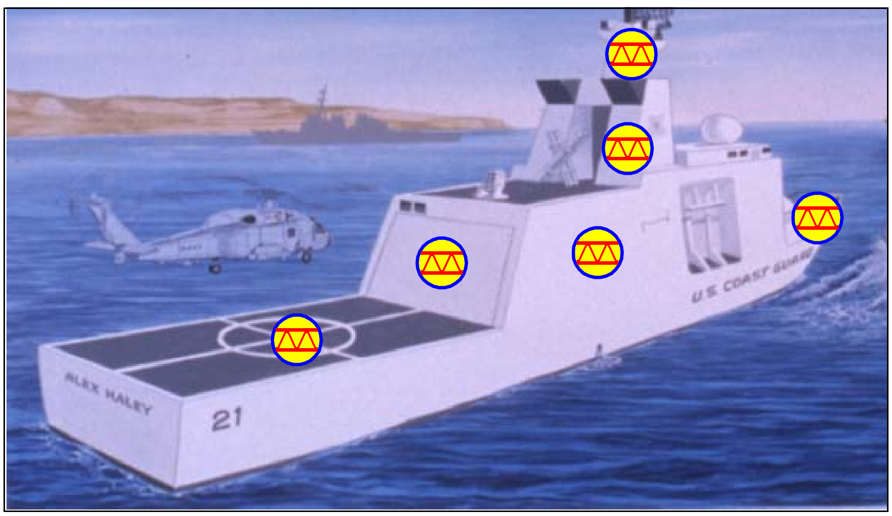


Figure 4.4.1-2: Proposed LASCOR Applications

Sandwich metal structures have a number of advantages over conventional steel construction:

1. Compared to conventional steel structures, metallic sandwich structures have reduced weight and increased stiffness. They are ideal for secondary structures such as internal decks, ramps, hatch covers, bulkheads, and deckhouses, with weight savings of 20 to 50 percent over conventional steel construction.
2. They result in reduced fabrication and outfitting costs. LASCOR panels are 20 percent cheaper to build and install than steel grillages. They have a high dimensional stability that helps reduce assembly and fit-up costs in the shipyard. Outfitting of distributive systems and installation of insulation costs are also reduced because of the smooth surfaces resulting from the elimination of most of the stiffeners.
3. The elimination of stiffeners on decks and bulkheads increases the usable volume within the total ship.

4. Metallic sandwich panels have improved thermal and fire performance. The space within the core offers inherent thermal insulation and protection against the spread of fires.
5. The high stiffness of sandwich panels reduces vibrations. Panels can also be sound-isolated from surrounding structures.
6. Laser-welded sandwich panels are ideal applications of automated fabrication techniques. They can be pre-fabricated as panels at high-efficiency factories before shipboard installation in the shipyard.

One of the issues with sandwich structures is corrosion protection of the voids within the core. Stainless steel applications, currently used in the fleet, are one solution. They have suffered no corrosion or fatigue damage after a decade of service. Another solution, which has been successfully tested in the field with ordinary steels (carbon steels), is to fill the void spaces with foam. Although our experience is limited to steels, sandwich panels can also be made from corrosion-tolerant metals such as aluminum or titanium. Such lightweight materials can further reduce structural weight and would be available for the mid to far-term applications.

4.4.1.2 Composite Structures

Composites have been used as the primary structure for small vessels for many years. They have also been used for secondary components. Below, in Figure 4.4.1-3, is a summary of the composite applications available.

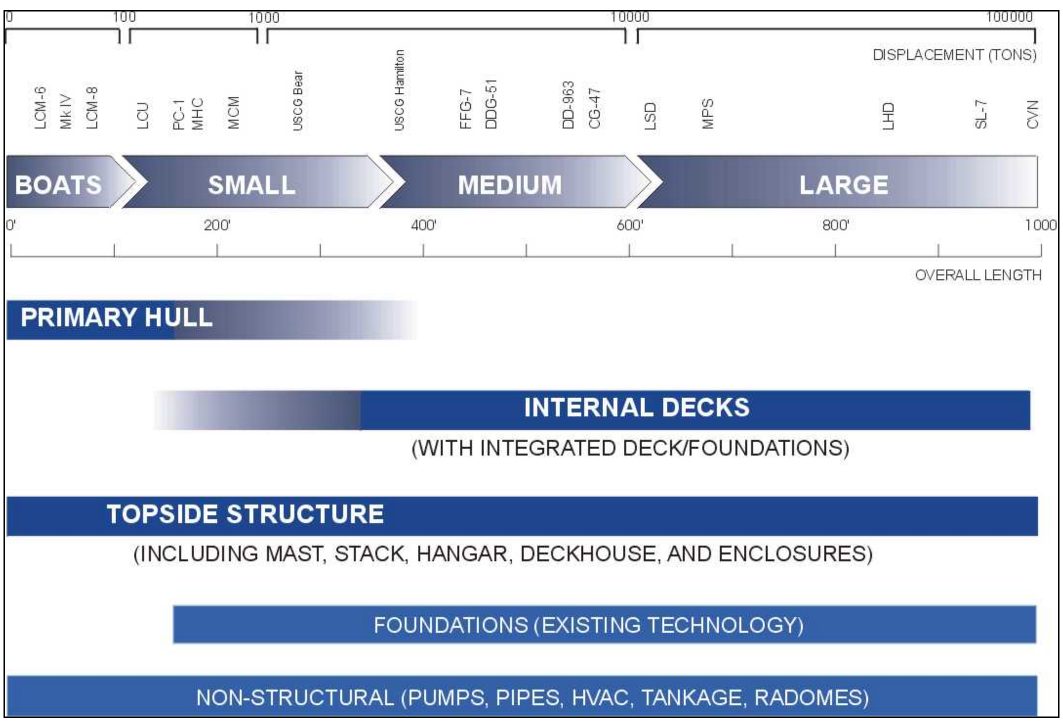


Figure 4.4.1-3: Composite Applications

Composites offer many advantages compared to standard metallic structures:

1. They are lightweight. Weight reductions of 35 to 50 percent, compared to steel, can currently be realized for secondary structures made of E-glass composite laminates. Since secondary structures comprise a significant fraction of the total structural weight, this translates into a total ship weight savings of about 8 percent of a large vessel (nominally 800 feet long).
2. Composite structural elements have better dimensional stability than steel elements. This is an aid to the fit-up and assembly in the shipyard, and results in lower fabrication costs and better overall dimensional tolerances.
3. They have reduced noise and vibration properties. Composites have inherently better damping and compliance than metallic structures. They also have the potential to be adapted into smart structures, i.e., structures that can monitor and/or alter their properties in service.
4. Fires are more easily contained in composite structures because of their low thermal conductivity. The cores in composite sandwich panels are good thermal insulators.
5. The designer has increased flexibility to tailor the composite structure to the particular need. Complex geometries can be designed to optimize the strength and stiffness, or to enhance producibility by minimizing the number or location of joints.
6. Composites have lower life-cycle maintenance costs than steel structures. Fewer inspections, less painting, and fewer repairs are needed over the life of the ship because of the non-corrosion and reduced fatigue damage of composites over metallic structures.

Tables 4.4.1-1 through 4.4.1-4 (Reichard, 1988)[1] present the relative weights of panels having equal stiffness and equal strength under both in-plane (axial) and bending loads. Composites are more advantageous than steel or aluminum when compared on an equivalent strength basis rather than on stiffness basis.

[1] Reichard, Ronnal, P., "Material Selection for Boats and Ships," Second International Conference Marine Applications of Composite Materials, Florida Institute of Technology, Melbourne FL, 21-23 March, 1988.

Table 4.4.1-1: Panels of Equal In-Plane Stiffness*

Material	Skin Thick. (inch)	Core Thick. (inch)	Elastic Modulus (ksi)	Weight (lb/sqft)
Steel	0.08	0	30,000	3.36
Aluminum	0.25	0	10,000	3.62
E-Glass (0,90)	1.14	0	2,200	9.99
Kevlar (0,90)	0.60	0	4,200	4.49
Carbon (0,90)	0.35	0	7,200	2.87
Uni-E-Glass	0.57	0	4,400	4.99
Uni-Kevlar	0.30	0	8,400	2.24
Uni-Carbon	0.17	0	14,400	1.43
E-Glass/Core (0,90)	0.57	5	2,200	15.15
Kevlar/Core (0,90)	0.30	3	4,200	7.59
Carbon/Core (0,90)	0.17	1.75	7,200	4.68
Uni-E-Glass/Core	0.28	3	4,400	8.09
Uni-Kevlar/Core	0.15	1.5	8,400	3.79
Uni-Carbon/Core	0.09	1	14,400	2.47

* All panels have a stiffness of 2.5×10^6 pounds/inch

Table 4.4.1-2: Panels of Equal In-Plane Strength*

Material	Skin Thick. (inch)	Core Thick. (inch)	Yield Strength (ksi)	Weight (lb/sqft)
Steel	0.19	0	80	7.56
Aluminum	0.26	0	58	3.74
E-Glass (0,90)	0.34	0	44	3.00
Kevlar (0,90)	0.25	0	60	1.89
Carbon (0,90)	0.14	0	105	1.18
Uni-E-Glass	0.17	0	88	1.50
Uni-Kevlar	0.13	0	120	0.94
Uni-Carbon	0.07	0	210	0.59
E-Glass/Core (0,90)	0.17	1.75	44	4.80
Kevlar/Core (0,90)	0.13	1.25	60	3.18
Carbon/Core (0,90)	0.07	0.75	105	1.96
Uni-E-Glass/Core	0.09	1	88	2.53
Uni-Kevlar/Core	0.06	0.5	120	1.46
Uni-Carbon/Core	0.04	0.5	210	1.11

* All panels have a maximum strength of 15.0×10^3 lbs/inch width

Table 4.4.1-3: Panels of Equal Flexural Stiffness*

Material	Skin Thick. (inch)	Core Thick. (inch)	Mom. of Inertia (inch^4)	Weight (lbs/sqft)
Steel	0.74	0	0.0335	29.74
Aluminum	1.06	0	0.1004	15.39
E-Glass (0,90)	1.76	0	0.4543	15.46
Kevlar (0,90)	1.42	0	0.2386	10.71
Carbon (0,90)	1.19	0	0.1387	9.80
Uni-E-Glass	1.40	0	0.2262	12.25
Uni-Kevlar	1.13	0	0.1196	8.51
Uni-Carbon	0.94	0	0.0697	7.79
E-Glass/Core (0,90)	0.23	2	0.4539	6.04
Kevlar/Core (0,90)	0.16	1.75	0.2380	4.15
Carbon/Core (0,90)	0.15	1.375	0.1385	3.83
Uni-E-Glass/Core	0.15	1.75	0.2272	4.41
Uni-Kevlar/Core	0.13	1.375	0.1185	3.31
Uni-Carbon/Core	0.11	1.125	0.0692	2.96

* All panels have a stiffness (EI) of 1.0×10^6 pound-inch^2

Table 4.4.1-4: Panels of Equal Flexural Strength*

Material	Skin Thick. (inch)	Core Thick. (inch)	Yield Strength (ksi)	Weight (lbs/sqft)
Steel	0.19	0	80	7.56
Aluminum	0.26	0	58	3.74
E-Glass (0,90)	0.34	0	44	3.00
Kevlar (0,90)	0.88	0	17	6.65
Carbon (0,90)	0.14	0	105	1.18
Uni-E-Glass	0.17	0	88	1.50
Uni-Kevlar	0.44	0	34	3.33
Uni-Carbon	0.07	0	210	0.59
E-Glass/Core (0,90)	0.12	1.25	44	3.47
Kevlar/Core (0,90)	0.20	2	17	5.09
Carbon/Core (0,90)	0.09	0.75	105	2.19
Uni-E-Glass/Core	0.09	0.875	88	2.46
Uni-Kevlar/Core	0.15	1.375	34	3.61
Uni-Carbon/Core	0.05	0.625	210	1.52

* All panels have a maximum moment capacity of 7.5×10^2 foot*pounds

There are a number of issues associated with composites:

1. Flammability, smoke, and toxicity dangers are the main concerns associated with composites. They are handled in several ways. For unmanned spaces in secondary structures not subject to severe fire threat, a thin thermal barrier coating, no coating, or passive fire protection may be used. For manned spaces in secondary or primary structure, thermal protective insulation is used (see Figure 4.4.1-4).

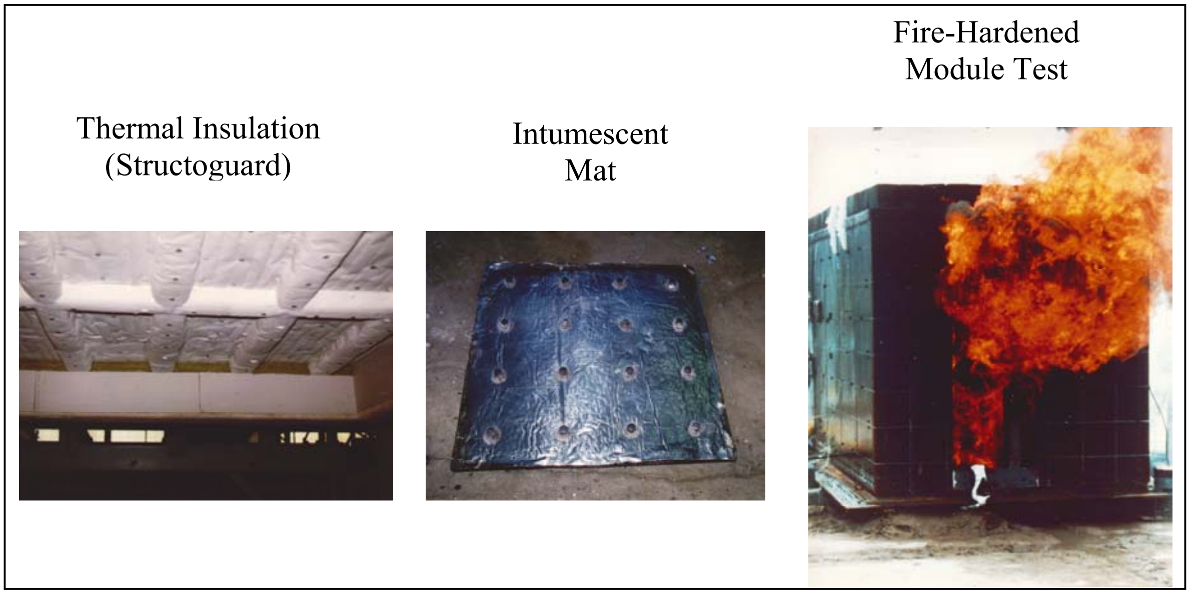


Figure 4.4.1-4: Composite Fire Protection

2. Composite designs are normally limited by stiffness, not strength. For hybrid structures, such as a composite deckhouse on a steel hull, the lower stiffness results in lower stresses and better fatigue performance. However, for the primary hull structure of large ships, the low hull overall stiffness may be a problem for deflections limits of conventional propeller shafts. Other propulsors (such as waterjets, electric drive, and podded propulsion) may render this issue moot.
3. There are limited design data and analytical tools. Design data and tools are becoming increasingly available for more common materials (e.g., glass polyester or vinylester) and for structural configurations, joints, and fabrication processes. However, in most cases, experimental validations are still needed.
4. There is minimal shipyard experience for constructing large composite ships. The largest composite ship hulls are those of naval minehunters and minesweepers, with lengths of 50 to 60 meters.

4.4.2 Technology Goals

There are a number of issues that must be explored before LASCOR and composite structural technologies are considered for high-speed sealift primary and secondary structural applications. Obviously, because LASCOR and composite structural concepts have been demonstrated for

secondary applications, the research necessary to implement them for high-speed sealift secondary applications is not as extensive as for primary structure applications, and could be accomplished in the near-term. There are several issues to be resolved before either LASCOR or composites can be considered for primary structural applications, hence, the earliest they could be available for primary structural consideration would be in the far-term applications.

4.4.2.1 LASCOR

Although LASCOR has been used in commercial and military secondary structural applications, several issues need to be addressed before it can be reliably used for high-speed sealift secondary and primary structures.

As mentioned earlier, one of the issues with metallic sandwich structures is corrosion protection of the voids within the core. As discussed, stainless steel has already been used in the fleet to eliminate this problem in several applications and has suffered no corrosion or fatigue damage after a decade of service. Another solution, which has been successfully tested in the field with ordinary steels (carbon steels), is to fill the void spaces with foam. These results have been promising. Our mid to far-term goal is to develop sandwich panels from lightweight, corrosion-tolerant metals such as aluminum or titanium and optimize the sandwich panel characteristics such as structural weight, acquisition costs and life-cycle costs.

Other issues associated with manufacturing will be resolved as the shipyard experience increases with metallic sandwich panels. Efficient repair procedures need to be further developed and optimized. Draft design guides and standards exist, but must be formally documented and approved by the Navy and regulators for commercial applications. The fatigue performance of metallic sandwich panels must be further defined and validated for both primary and secondary loads along with full-scale structural static tests. This will allow a reduction in the factors of safety now assumed, resulting in lighter and more reliable structures.

Particular to sealift applications, cargo decks must be optimized for specific vehicle loadings, ensuring the necessary ruggedness requirements without imposing an unnecessary weight penalty.

In the far-term, techniques to form complex shapes, not just flat or singly-curved panels, must be developed and optimized. The ability to form hybrid metallic sandwich structures also has far-term potential for weight and cost reductions.

4.4.2.2 Composite Structures

Similar to LASCOR, composites have been used in commercial and military secondary structural applications, but several issues need to be addressed before they can be reliably used for high-speed sealift secondary and large primary structures.

Although high-speed sealift requirements differ significantly from that of combatants, the Norwegian combatants *Oskoy* and *KNM Skjold,* and the Swedish combatants *Smyge* and *Visby* have demonstrated composite technological advancements for high-speed sealift applications.

Advanced composite materials and integrated structural systems were used for these hullforms, in part, for their ability to improve the signature characteristics and shock resistance of the structure. In addition, the reduced weight and simplified construction and outfitting demonstrated during the construction of these small vessels are important attributes in a high-speed sealift platform.

The recently-commissioned fast patrol craft *KNM Skjold*[2] (see Figure 4.4.2-1) is an SES hullform with an overall length of 47 meters and displacement of 260 tons. Fiber-Reinforced Plastics (FRP) sandwich construction is used throughout the vessel, with vinyl ester or polyester resins and either a PVC or PMI core material. In locations requiring high stiffness, carbon fibers were used; otherwise, E-glass was chosen for the laminates.

The Swedish corvette *Visby*[3] (see Figure 4.4.2-1) is also one of the more advanced combatants using composite technology. With an overall length of 73 meters, the *Visby* is the largest commercial or combatant ship hull entirely constructed of Carbon Fiber-Reinforced Plastics (CFRP). The *Visby* was built using a vacuum-infused process consisting of sandwich construction with a PVC foam core and vinyl ester resin.

These small warships demonstrate the flexibility and capability of composite technologies for small surface combatants. Further research to improve material properties, joints and connections, and shipyard producibility is necessary before these technologies are matured for primary hull structures of larger ships.

Skjold Visby

Figure 4.4.2-1: Composite Small Combatants

Research has shown the potential of multi-layered, balsa-cored sandwich structures for containing fires and preventing structural collapse or excessive deflections. The French surface combatant *La Fayette* uses balsa-cored sandwich construction on the deckhouse for this purpose. Also, the potential use of phenolics and other fire-retardant resins has been demonstrated for

[2] http://www.knmskjold.org/english, 2000.
[3] http://www.naval-technology.com/projects/visby, 2001.

small fires burning for 20 to 30 minutes. These potential fire performance improvements must be further developed.

Although joining technology is very critical for all composite structural applications, it is particularly important when considering composite materials for primary structure. There is a significant reduction in in-plane strength characteristics at the joints of composite structures. This problem can be eliminated or reduced if a monolithic rather than modular construction process is adopted. However, for very large ship lengths, it would seem that modular construction would be required, and a significant research and development effort would be needed to improve composite joining technology.

There are limited design data and analytical tools available. Design data and tools are becoming increasingly available for more common materials (e.g., glass polyester or vinylester) and for structural configurations, joints, and fabrication processes. However, in most cases, further development and experimental validations are still needed.

Inspection and repair methods are generally available for most composite structures. However, inspection can become difficult for thick, sandwich structures. Therefore, some of the current non-destructive evaluation (NDE) methods must be further developed.

4.4.3 Projected Weight Savings

Table 4.4.3-1 is a summary of the projected weight savings for the various materials in the near, mid, and far-terms. All of the weight savings are relative to ordinary steel (ABS grade A) of conventional stiffened plate construction. The percentage reductions are applied to the structural weight (SWBS 100) of the entire ship.

Table 4.4.3-1: Summary of Weight Savings (Percent)

Material	Near-Term	Mid-Term	Far-Term
Aluminum	30	30 - 40 with new alloys	50 with improved joining technologies
Titanium & Advanced Metals	High Risk	40 - 55 secondary structure, 15 overall	20 - 60 overall
Metal Sandwich (LASCOR)	35 - 50 secondary structure, 10 overall (steel)	40 - 55 secondary structure, 15 overall (steel)	45 - 60 secondary struct, 20 - 30 overall (hybrid metals)
Composites (300' ship length)	20 - 40 with Glass or Carbon fibers	30 - 45 with Glass or Carbon fibers	35 - >65 with new fibers & resins
Composites (800' ship length)	8 with Glass or Carbon fibers	35 - 45 with Glass or Carbon fibers	50 - >65 with new fibers & resins

4.4.4 Development Plan for Structures

The high-speed sealift missions require not only the further development of low-cost, high-strength/lightweight materials, but also the most cost and weight efficient structural concepts that can implement these material enhancements. For secondary structures and for the primary structure of ships less than 130 meters, several lightweight/relatively low-stiffness materials are already being used in particularly weight-critical applications. Cost considerations often dictate the use of steel construction for components that are not weight-critical.

Sandwich metal structures such as LASCOR have been used for non primary load carrying structure and have been effective in reducing structural weight. To optimize LASCOR secondary structures and strength decks for high-speed sealift applications, additional research and development would be needed. In the near-term, these efforts would include:

- The development of optimum ways to reduce or eliminate corrosion within the void spaces such as using alternate lightweight metals as the primary material or filling the void spaces with foam.
- The further development of efficient repair procedures and inspection techniques.
- The documentation and approval of design guidelines.
- Testing and evaluating primary load carrying capacities and fatigue performance.
- The optimization of cargo decks.

In the far-term, techniques to form complex shapes, not just flat or singly-curved panels, must be developed and optimized. Large-scale tests would be needed to determine LASCOR performance for primary load carrying applications. This would be a significant research and development effort. The ability to form hybrid metallic sandwich structures also has far-term potential for weight and cost reductions.

To optimize composite secondary structures and strength decks for high-speed sealift applications, additional research and development would also be needed in addition to the material developmental effort described earlier. These efforts include:

- Significant testing, analysis, documentation, development of design tools, and approval of design guidelines.
- Joint detail development.
- The further development of efficient repair procedures and inspection techniques.
- Testing and evaluating primary load carrying capacities.
- The development of cores and materials for fire containment.
- The optimization of cargo decks.

Although composites have been used for primary structural applications in vessels below 130 meters, there is no experience with its use for large vessels, and a significant development effort would be required. Because of the extensive research needed before composite structures can be used as the primary hull material for very large ships, it is considered a far-term technology goal. In addition to the model tests discussed earlier to determine loads, full-scale demonstrations verifying large-scale joining technologies, manufacturing processes, and at-sea performance would be needed.

4.4.5 Summary of Required Technology Development

Table 4.4.5-1 is a summary of the technologies to be investigated and developed for use in the near, mid, and far-term high-speed sealift structures.

Table 4.4.5-1: Technology Development Needs

Material	Near-Term	Mid-Term	Far-Term
Aluminum	State-of-the-Art for ship lengths < 300', untried >300'	new alloys with better fatigue properties	improved joining technology for better fatigue properties
Titanium & Advanced Metals	no time for multi-year R&D effort	define fatigue & strength properties	improved joining tech. for shipyards
Metal Sandwich (LASCOR)	approved design standards & rules (steel)	validate fatigue properties & improve corrosion resistance	form complex shapes & develop hybrid metal applications
Composites (300' ship length)	State-of-the-Art for ship lengths < 200', better fire resistance	improved & validated design tools	develop high strength fibers & resins
Composites (800' ship length)	improve fabrication methods	improve inspection & repair methods	experimental validations

In addition, there are a number of other technologies that may become available for far-term applications. These technology thrusts include smart structures, adaptive structures, fiber placement/resin infusion, automated welding/joining, modular vessel components, and improved analytical/design methods. They are not only being pursued to reduce structural weight in future ships, but may also be effective in improving performance, enhancing fatigue and corrosion properties, reducing costs, and improving system reliability. Limited work is underway in these technologies around the world and in different industries, but it is not directed toward high-speed transport ships. The industries developing such technologies include aerospace, transportation, infrastructure, electronics, and offshore platforms. A comprehensive approach is needed to develop and transition these technologies to fast sealift ships in the far-term.

4.5 Preparation of ABS Guide for HSS

Classification is the process of verifying that the hull, machinery, and electrical systems and related components meet technical requirements for fitness, safety, and environmental soundness. These technical requirements are contained in Rules that are developed by the classification society. The vessels are verified to comply with rule requirements in their original design plans, as constructed, and throughout their operational life. The set of Rules to which a vessel is designed varies depending on its type of classification and service, as well as any special notations; for example, most large cargo vessels and large passenger vessels are Classed under Steel Vessel Rules, while many high-speed ferries may be Classed under High-Speed Craft Rules. ABS is currently in the process of developing *Rules for Building and Classing Naval Vessels* and *The ABS Guide for Building and Classing High Speed Naval Craft* for warships and vessels engaged in military missions.

Current sealift vessels are built and maintained to ABS class using the commercial *ABS Rules for Building and Classing Steel Vessels,* with enhancements identified as necessary to support the military mission specified in the acquisition contract. If high-speed sealift vessels are procured, it would be possible to follow the same philosophy with the additional requirements called out in the *ABS Guide for Building and Classing High Speed Craft*. However, applicability of the rules to current mission profiles being considered for these high-speed sealift concepts will have to be considered, the far-term materials and joining technologies being developed for these missions will have to undergo the certification process, and, as described earlier, further data is needed on the large mono and multi-hull variants being considered that currently fall outside our experience base.

A process was developed to ensure that past experience would not be lost in the development of Naval Vessel Rules. This process included a comparison of naval and commercial standards that led to an initial draft set of standards. These draft standards were then reviewed and modified by technical committees and industry, but have not been reviewed or approved by the Navy. Finally, provisions were made for annual updates of the standards.

A similar process is envisioned for the development of the HSS rules, with a few exceptions. At the current time, the industry does not have any experience designing the hullforms being considered for the high-speed sealift missions. Therefore, to prepare for an initial draft set of standards, significant research as described previously in loads, materials and high-strength/lightweight structures development is necessary. Because a set of standards ensuring the fitness, safety, and environmental soundness for high-speed sealift vessels is the ultimate goal of this program, ABS involvement in the research and developmental efforts is planned to ensure that they will be provided with all the necessary information to class the vessels.

4.5.1 Surveys After Construction

Though a new vessel may be granted classification and thereby judged fit for its intended service, such status is not automatically retained throughout its service life. As the rigors of sea can be wearing on a vessel's hull and machinery, the society conducts periodic surveys to determine whether a vessel is being maintained in a condition worthy of retaining classification

status. This is particularly important for the high-speed sealift ships. The lack of experience with the unique structures and materials being considered for high-speed sealift variants, in conjunction with speeds outside typical displacement hullform parameters, may dictate a rigorous inspection plan.

5.0 MACHINERY SYSTEMS

5.1 Introduction

Significant extension of machinery technology is required to meet the needs of HSS ships. Propulsion machinery must be compact, lightweight, and fuel-efficient, yet produce and transmit very high levels of power. The technical experts at the HSS Technology Workshop identified the types of propulsion machinery components (gas turbines, reduction gears, and waterjets) to meet HSS needs. However, power, weight, and efficiency requirements exceed current capabilities for each of these components, particularly for the far-term ship concepts.

5.2 Prime Movers

Prime movers for HSS designs range in power from small current technology turbines producing about 10 MW, to large far-term turbines producing about 100 MW. Existing gas turbines with ratings of up to 30 MW are adequate for a number of the HSS missions, particularly those intra-theater missions with limited range, lower speed, and modest cargo. Progressively more powerful, more fuel-efficient turbines are needed as speed, range, and payload increase. The designs show requirements for a near-term nominal 43 MW turbine and a far-term nominal 90 MW turbine. While fuel efficiency is important for all turbines, it is particularly important for the 90 MW turbine since these large turbines are associated with the higher speed, long-range inter-theater missions. Significant gas turbine technology development is required to meet power and fuel consumption goals for missions requiring large near and far-term technology turbines.

5.2.1 State-of-the-Art

The General Electric LM2500 gas turbine, at a Navy-certified maximum continuous power rating of 26,250 HP (100^{o}F MCP), is widely used in a large number of U.S. Navy ships. The LM2500 is an aero-derivative gas turbine that was directly derived from GE's CF6 family of commercial aircraft engines and GE's TF39 military engine. This engine, in industrial applications, operates continuously at up to 33,000 HP and is certified by at least two foreign Navies at 29,000 HP. GE estimates that this engine could be Navy-certified at 30,000 HP with a Mean Time Between Repair (MTBR) of 2,000 hours. There are over 800 LM2500 gas turbines in service in more than 24 international navies.

GE LM2500+ turbines are currently in service powering waterjets in the 42-knot ferry Corsaire 13000 Liamone with the turbine rated at 25 MW. The turbine is also in service in the cruise liner Millennium in an integrated electric plant. The first military application of the LM2500+ will be in the LHD Wasp-class large-deck multi-purpose amphibious assault ship. The LM2500+ will have a U.S. Navy rating of 26.1 MW (35,000 shp) for the LHD application. U.S. Navy certification at this rating is planned as part of this shipbuilding program.

The GE LM5000 and the United Technologies (Pratt & Whitney) FT-9 and FT-8 gas turbines are nominal 30 MW (40,000 shp) gas turbines. The LM5000 was first introduced in 1982 for industrial applications and over 100 units have been placed in operation. GE maintains that the

LM5000 could be readily Navy-certified at approximately 28 MW (38,000 shp). The LM5000 is expected to have about a 5 percent improvement in efficiency compared to the LM2500. The FT-9 was developed under Navy sponsorship and was Navy-Certified at 25 MW (33,000 shp) in 1980. The FT-8 is currently under development. The FT-8 is physically smaller than the FT-4 and is designed to operate on DFM fuel. The capability to produce marine versions of the FT-9 and FT-8 currently exists at Turbo Power and Marine Systems, Inc. (TPMS) (a subsidiary of United Technologies) in the United States. Marine versions of the LM5000 and FT-8 could both be produced, tested and certified in approximately four years at a cost of approximately $100 M each.

Gas turbine R&D advances have resulted in some simple-cycle plants operating with efficiencies of more than 40 percent at full power, but with significant reductions in fuel efficiency at part power. Turbines using more complex cycles exploiting intercooling and recuperation (ICR) technologies reportedly achieve specific fuel consumption rates closely approaching the very flat curve characteristic of larger diesel engines. Warships may be the first to benefit from ICR technology in the form of the ICR-based Northrop Grumman/Rolls Royce 25 MW WR21 marine gas turbine, an engine derived from the aero engines (RB211 and Trent) combined with intercooler and recuperator systems. This engine has successfully completed a 500-hour land-based endurance test in a joint US/UK/France program. The WR21 is more fuel efficient than simple-cycle gas turbines across its entire power range. A maximum reduction of about 30 percent is achieved at the bottom of the power range, with savings of about a quarter that at full power. The WR21 turbine has been ordered for use in the UK Type 45 destroyer in 2007. The size and complexity of the WR21 ICR gas turbine and the LM2500 turbine are illustrated in Figure 5.2.1-1.

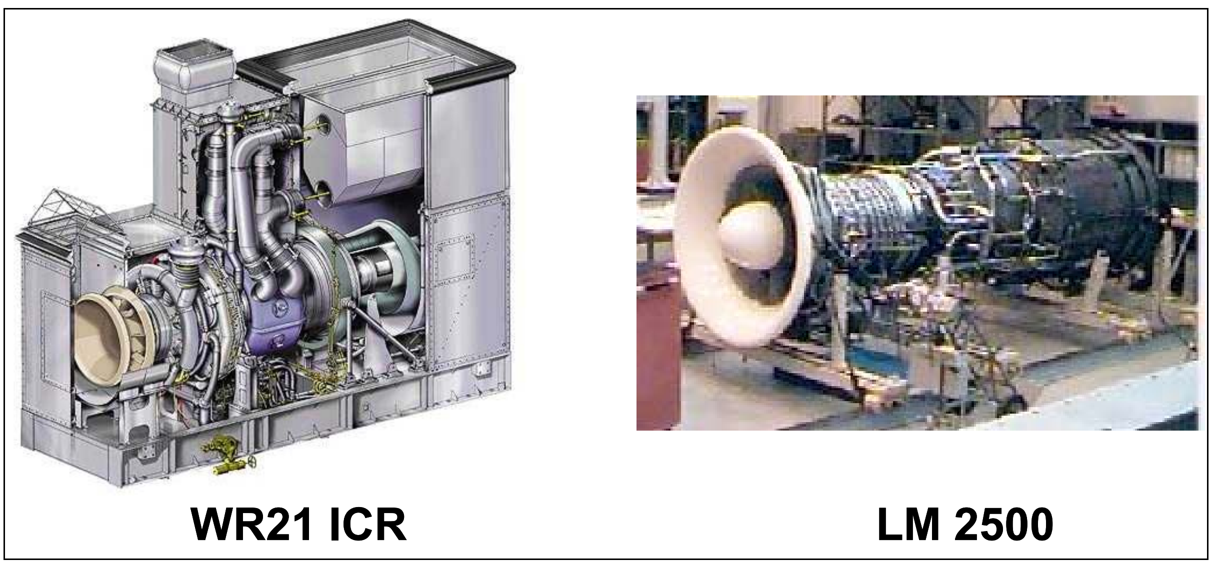


Figure 5.2.1-1: WR21 ICR and LM2500 Gas Turbines

GE Marine Engines' uprated LM6000 aero-derivative industrial gas turbine is another developmental choice for marine propulsion application. Although the LM6000 is designed for offshore use, environmental requirements are similar to those of marinized turbines. The major

development issue would be modifications to the control system to be compatible with propulsor loading requirements. The LM6000 uprated models offer over 40 MW (50,000 shp) with 42 percent ISO thermal efficiency. Currently, there are 115 LM6000s in operation. GE is considering development of a version of the LM6000 for the offshore industry with a free-power turbine and intercooling in the range of 73-77 MW, thermal efficiency of 43.9 percent, and a specific fuel consumption of 191 g/kwh.

The development of the FastShip Atlantic project resulted in a design for a 40-knot, 1420 TEU commercial vessel powered by five 50 MW (68,000 bhp) gas turbines. Machinery proposals were submitted by GE and Rolls Royce. GE Marine offered a marine version of its industrial 42 MW (57,100 bhp) LM6000, while Rolls Royce offered the 47.5 MW (64,600 bhp) Marine Trent derived from its aero 800 Trent turbine. While both are untried in the marine environment, they were found to be technically acceptable for the project. The Trent was selected to power the FastShip design, although construction has not begun. Full-power simple-cycle efficiency is 42 percent for the turbine.

GE, with DOE support, has started development of the LM9000, a 75^{+} MW engine for industrial use. The new turbine would be developed from an existing aero-derivative core for power generation use starting around 2008. Use of a GE90 core for the LM9000 could result in a maximum power of 125 MW, while a CF6-80C2 core could result in a maximum power of 90 MW. Preliminary estimates for power, module size, and area and velocity for both inlet and exhaust make it attractive for marine use.

The specific fuel consumption rate and power of existing and developmental gas turbines is compared with HSS near and far-term goals in Figure 5.2.1-2. The figure shows that near-term goals can largely be met by marinized LM6000/Trent engines. Far-term gas turbine objectives require development of a turbine like the LM9000 to meet power objectives, as well as significant reduction in fuel consumption.

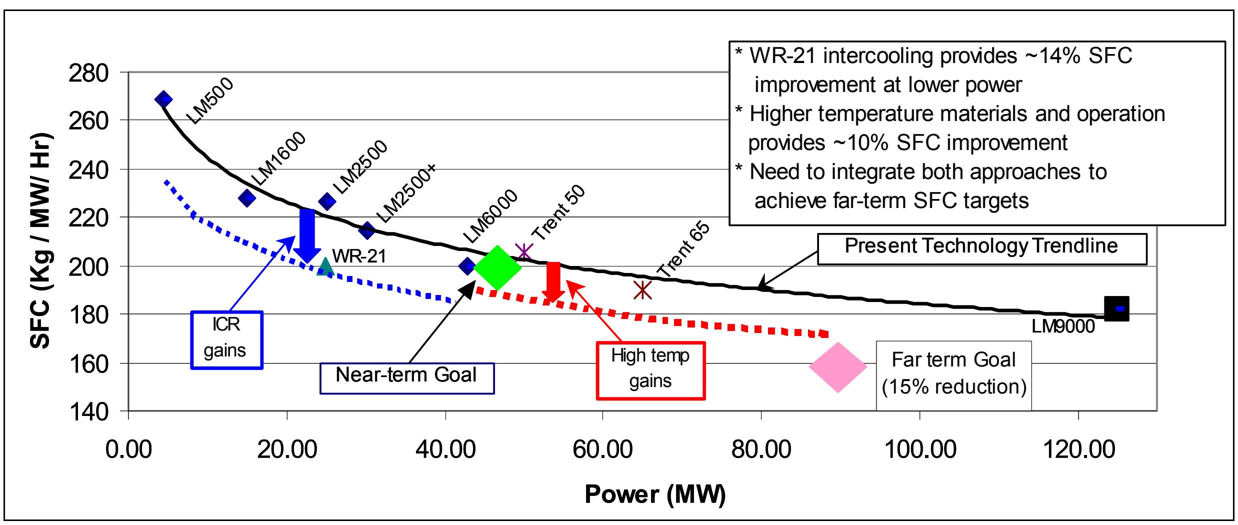


Figure 5.2.1-2: Marine Gas Turbine Technology

5.2.2 Technology Goals

The near-term gas turbine technology goal is development of a marinized 43 MW gas turbine with SFC of 200 g/kWh, very similar to the turbines required for the FastShip Atlantic project. Such a gas turbine has essentially the same turbine performance requirements as for the commercial FastShip Atlantic project. The FastShip Atlantic project experience indicates that the commercial approach to achieve this objective is to develop marinized versions of the LM6000 or Rolls Royce Trent industrial turbines. Development of a near-term turbine requires 3-4 years.

The far-term gas turbine technology goal is development of a marinized 90 MW gas turbine with SFC of 158 g/kWh. Several options exist for developing turbines with the required power in this power range, including marinizing and enhancing the industrial LM6000, combining WR21 ICR technology with a more powerful core, or developing the LM9000 marine turbine. Meeting far-term turbine specific fuel consumption goals will require a combination of technologies, including improved high-temperature materials, advanced gas turbine blade technology, and intercooling and recuperator systems (similar to those used on the WR21 ICR).

Commercial market forces may lead to development of an industrial LM6000 with a free-power turbine and intercooling that can produce approximately 73-77 MW of power. The thermal efficiency would be in the range of 43.9 percent, with an SFC of 191 g/kwh. Although falling short of the far-term turbine power and fuel consumption goals, the base of commercial interest in the turbine may make it significantly more affordable than alternatives that fully meet the goals. This turbine could be developed in about 3 years.

Far-term gas turbine requirements could be reached by combining technology developed for the WR21 ICR project with higher power cores. Use of the Trent 900 core would result in a nominal 90 MW engine. The 90 MW WR21 could be developed in about 6 years.

The LM9000, depending on the core used, could develop power ranging from 75 MW to 125 MW. Using the GE90 core would result in a maximum power of 125 MW. A CF6-80C2 core could result in a maximum power of 90 MW. GE estimates the 75 MW range is probably a more reasonable power level. Development of the LM9000 to meet HSS power requirements would take about 3-4 years. Development of the technology to reach fuel consumption goals requires an additional 3-4 years.

In addition to these specific engine-oriented development approaches, there are more generic gas turbine technology development efforts focused on key technologies that may indirectly support HSS objectives. In the U.S., DoD, NASA, and the U.S. aerospace industry jointly fund the Integrated High Performance Turbine Engine Technology (IHPTET) with the objective of developing and demonstrating advanced engine technologies. IHPTET is producing revolutionary advancements in turbine engine technologies due to the synergistic effect of combining advanced material developments, innovative structural designs, and improved aero-thermodynamics. Recent accomplishments of the program include turbofan and turbojet designs now being developed that can achieve a 40 percent increase in thrust-to-weight and a 20 percent reduction in fuel consumption over baseline engines. Japanese research collaboration between

Ishikawajima, Kawasaki, Daihatu, Niigata and Yammen was initiated in 1997 to develop super industrial land-based gas turbines with improved emissions and specific fuel consumption. A different Japanese research effort is aimed at development of a large, highly efficient, gas turbine using heat-resistant ceramic compounds rated at 1700^{o}C. It is expected the higher operating temperature of the new gas turbine will improve fuel efficiency by 10 percent. While these independently-funded efforts are not directed at the HSS goals, the resulting materials, components, and technologies developed may benefit future marine turbines.

5.2.3 Overview of Development Plan

Gas turbine technology development is required to produce marine engines with the increased power and improved specific fuel consumption required for far-term HSS missions. While HSS power objectives have been met by aero and industrial engines, these high-power engines lack essential features needed for marine application (compactness, light weight, able to operate in a marine environment, etc). Further, technology enhancements are required to meet the fuel efficiency requirements for far-term HSS application. Development is required in two major areas to achieve the target goals – increased power and lower specific fuel consumption.

Development of marine gas turbines with the required power is simpler due to the existence of previously-developed industrial, offshore, and aero engines. Existing engines such as the LM5000, LM6000, and Rolls Royce Trent produce adequate power for near-term 43 MW turbine requirements, but may require marinization. Market forces such as the FastShip Atlantic project may lead to commercial development of these marine turbines independent of military investment. Full-scale fabrication and testing is not required since most of the units have been installed on land or offshore bases and have many hours of operation. Only the marinization of these units and modification to the control of the units is required.

Development of more powerful 75-100 MW far-term turbines for the marine industry will benefit from the cores of existing aerospace turbines such as GE's 90 MW CF6-80C2 and 125 MW GE90. Commercial market forces in the power generation and industrial industries may lead to development of industrial power generation turbines without military investment. However, military investment is likely to be necessary for marinization of these far-term turbines and development of technology enhancements to achieve fuel efficiency improvements.

Improvement of gas turbine specific fuel consumption can be realized through two approaches. One is through development of high-temperature materials for the gas turbine, along with gas turbine technologies like advanced blade designs. Studies indicate that this approach would yield a reduction of approximately 10-12 percent, marginally less than the far-term objective. The other approach is through development of intercooling and recuperator (ICR) systems for the larger engines. Projections for the WR21 ICR system currently being developed in the 25 MW power range indicate that use of similar technologies to the larger engines would meet far-term HSS requirements.

The tasks, time to complete each task, and costs associated with developing the needed gas turbine technology are shown in Figure 5.2.3-1. Two stages of gas turbine development are

shown to address near and far-term requirements. Costs shown are engineering estimates, based on the expected scope of testing and facilities required.

Time to complete (years)	1	2	3	4	5	6	7	8	9	10	Funding ($K)
Identify Power Concepts											500
Near Term Power (43-50 MW)											
- Engineering Design											62,500
- Detail Engineering, Material											125,000
- Assembly, Testing											62,500
Far Trem Power (90+ MW)											
- Engine Marinization											
- Engineering Design											87,500
- Detail Engineering, Material											175,000
- Assembly, Testing											87,500
- SFC Improvements											
- Identify SFC Concepts											5,000
- Engineering Development											200,000
- Assembly, Testing											70,000
- Integration, Testing											35,000
Funding ($K)	2,750	165,250	225,000	97,500	122,500	192,500	105,000				910,500

Figure 5.2.3-1: Marine Gas Turbine Technology Development Plan

5.3 Waterjets

5.3.1 Introduction

Waterjet propulsion is the preferred propulsion system for large, very high-speed HSS vessels. To date, the market for waterjets has been dominated by small ships and fast ferries. Large, high-speed ocean-going vessels will require large amounts of power to be transmitted to the waterjets for propulsion. To keep the number of waterjets per vessel to a reasonable number, waterjets that can absorb up to several times the maximum power of today's most powerful waterjets will be required.

5.3.2 State-of-the-Art

Large waterjets have basically followed two design approaches; one with mixed-flow and the other with an axial-inducer type blading design. The two primary makers of large waterjet units are KaMeWa and John Crane-Lips. Both of these makers have a long background in the waterjet field and have designs based on non-inducer, mixed-flow type blading.

KaMeWa's largest operational unit is the size 180, with an impeller inlet diameter of 180 centimeters (5.9 feet), shown in Figure 5.3.2-1. Introduction of the 200 cm size 200 waterjet is imminent. These waterjets can be powered by an LM2500 size gas turbine, putting them in the 25 MW power range. The KaMeWa impeller is a mixed-flow design. Mixed-flow waterjets have radial growth of the blade tip through the impeller and produce a significant radial exit component of velocity. The maximum impeller diameter can be as much as 40 percent larger than the impeller inlet diameter. The exit housing has a bowl shape beyond this exit blading to

accommodate the stator blading and results in installation diameters that are 65-85 percent larger than the inlet diameter.

Figure 5.3.2-1: KAMEWA 180 Waterjet

KaMeWa has performed design work on a size 325 unit (inlet diameter of 325 centimeters, or 10.66 feet) under contract to meet FastShip Atlantic operating requirements of about 40 knots. The size 325 unit would absorb power in the range of 49 MW using a Rolls Royce Marine Trent, but no units have been built to date.

John Crane-Lips has built mixed-flow waterjets that cover up to the 7.5 MW range.

Rolls Royce/Bird-Johnson is developing the AWJ21 waterjet unit concept. This unit has an advanced mixed-flow impeller mounted in a nacelle arrangement that would be faired with the bottom of the ship hull and incorporates an underwater discharge. The waterjet steering/ reversing equipment would be housed within the nacelle for minimum drag impact. This unit is only in the model development phase, but is intended for up to LM2500 size power applications. The drag of the nacelle for the AWJ21 waterjet concept would probably not favor this approach for application on extremely high-speed ships, but would limit it to speeds of 40 knots and below.

Axial-inducer type waterjets have been developed and used on such ships as the Jetfoil, SES-100A, and PHM. The Jetfoil waterjets used single-stage axial inducers of 51 cm diameter that absorbed about 3.2 MW and were developed by Rocketdyne. The PHM units were developed by Aerojet Liquid Rocket Company and each had a power rating of 13.4 MW with a two-stage, two-speed axial-inducer waterjet of nearly 117 cm diameter. The SES-100A also had two Aerojet two-stage, two-speed axial-inducer waterjets rated at 6 MW each, which gave that vehicle speeds approaching 80 knots. Aerojet did extensive work for the 3KSES program on developing a 117 cm diameter, two-stage, single-speed axial-inducer pump of 30 MW intended for speeds of 90 knots. These units had inducer stage inlet tip speeds as high as 60 meters per second, with the high tip speed enabling reduced unit size. However, the 3KSES program was discontinued prior to full-scale testing of the hardware.

The Marine Corps AAAV program uses 58 cm single-stage axial-inducer waterjets of about .97 MW for water propulsion as a current application of axial-inducer waterjet technology. These units were developed by Naval Surface Warfare Center, Carderock Division.

Two-stage, two-speed axial-inducer waterjets were developed for the SES-100A and the PHM. The second stage in each pump turned at much higher RPM and tip speeds than the initial inducer stage. The headrise produced by the initial inducer stage permitted running much higher tip speeds on the second stage. Second stage tip speeds in excess of 90 meters per second were used. The advantages of the higher speed second stage are that it can be made shorter to save space and weight, and the operating point for the second stage can be in a more favorable flow coefficient range for stage hydraulic efficiency. Gearing and shafting is more complicated for two-stage, two-speed pumps.

Axial-inducer waterjets have seen fewer large-scale applications compared to mixed-flow designs. This is due mainly to the significant contraction in the large waterjet market that followed cancellation of the 3KSES program. Proponents of axial-inducer waterjet technology, such as Rocketdyne and Aerojet, discontinued their waterjet businesses about this time. Mixed-flow waterjet manufacturers remained in business by producing smaller units during this waterjet recession and subsequently were able to develop progressively larger mixed-flow waterjets as market demand evolved over the following decades. The belief that inducers were only applicable to low flow coefficients, and historical data that indicates that the mixed-flow pumps have a hydraulic efficiency advantage over axial designs, also led to the emphasis on mixed-flow designs by other waterjet manufacturers. However, the ability of axial-inducer type pumps to operate over a wider range of flow coefficients has been demonstrated, and any efficiency differences on design are expected to be slight. The high suction specific speed operational ability of the inducer type pump and the straight-through flow design results in much smaller, lighter, and faster turning designs than for other pump types. Consequently, axial-inducer pumps are more compact systems with lighter gearing. The installation diameter of an axial-inducer waterjet is only about 20 percent greater than its inlet diameter, while the installation diameter of a mixed-flow design is 65 to 85 percent greater than its inlet diameter. The smaller size is particularly important for installation in the restricted transom space available in the slender hulls of high-speed ships.

The power rating and design speed of existing and developmental waterjets is compared with the pumps needed for HSS near and far-term designs in Figure 5.3.2-2. The figure shows that both near and far-term power goals significantly exceed the power capacity of existing waterjets. Furthermore, while some near-term HSS designs are compatible with mixed-flow waterjets, many near-term and all far-term term designs require the reduced diameter of single-stage or multi-stage axial pumps to fit machinery in the slender hulls.

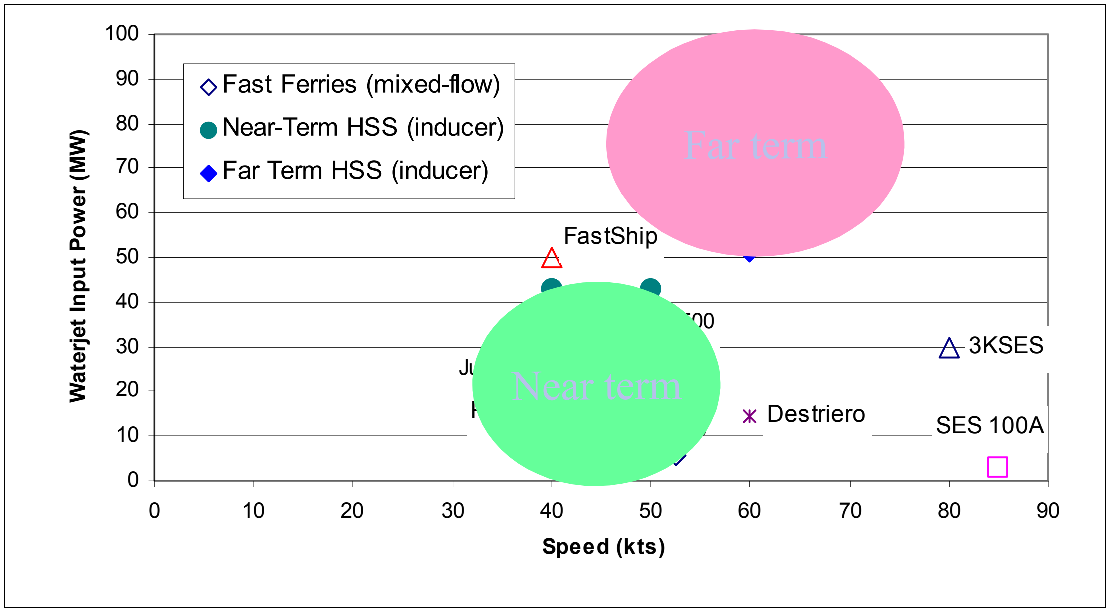


Figure 5.3.2-2: Waterjet Technology

5.3.3 Technology Goals

The technology to build waterjets that are matched to the high-power gas turbines and high speeds of HSS ships needs to be developed. The preferred configuration for HSS ships is one gas turbine for each waterjet. While demonstrated technology for mixed-flow waterjets is 25 MW, the power of axial-flow pumps similar to those needed for HSS applications has been limited to 10-13 MW. However, much of the design and manufacturing technology supporting today's more powerful mixed-flow pumps is shared by axial-flow pumps. The near-term goal is to extend the capability to manufacture axial-flow waterjets to ship speeds up to 50 knots with applied powers of as much as 43 MW. Meeting far-term goals requires extending to speeds as high as 70 knots and powers approaching 100 MW.

Single-stage axial-flow pumps were adequate for all of the HSS displacement hull designs and some of the SES designs. However, two-stage axial inducers are required for some SES designs where headrise requirements exceed the headrise ability of a single stage. This occurs for very high ship speeds and/or very high design point jet velocity ratios. Although two-stage, two-speed axial-flow waterjets have been built, HSS pumps require only single-speed pumps. The additional blade row is expected to be a somewhat standard design that does not involve any undue mechanical complication as it would co-rotate with the first-stage inducer and be driven with the same shaft. The two-stage design reduces the required tip speeds compared with a single-stage inducer design since the headrise per stage is reduced. However, for a comparable design point condition, both must pump a comparable flow rate and the two-stage units provide only a slight reduction in inlet diameter. Thus, the two-stage axial-inducer design is considered an extension of the single-stage design.

Inlet design will require development for the large, high-speed applications. Inlet design is not only tied to the waterjet design, but must be matched to the constraints of the high-speed hull design that is used. The long slender hulls, favored for high ship speeds, would limit the placement of inlets, and revised approaches to inlet design will need to be developed. The use of a single inlet to feed more than one waterjet is likely to be needed. The influence of hull boundary layer flow ingestion by the inlet has a large impact on waterjet performance. Waterjet inlets that maximize the capture of low-momentum boundary layer flow without adverse effects on ship drag will be favored. The impact of this low-momentum inlet flow can be as much as a 10-point improvement in the ship's overall propulsive performance coefficient. Ingestion of the lower-momentum boundary layer flow on the hull by the waterjet inlet will enable propulsive coefficients in the 70 percent range for very high ship design speeds. Such a performance improvement is an important design consideration for high-speed, long-range ships since it will have major impacts on the amount of power installed and the weight of fuel that the ship must carry. The nozzle location and orientation can generate additional lift by acting in a trim-tab like fashion to produce lift forces comparable to the total thrust of the jet. Understanding these effects and exploiting them to fully integrate hulls and propulsors offers significant potential for enhancing hydrodynamic performance. Inlet, hull, and jet interactions and inlet design can be best studied and developed through the use of computational fluid dynamics (CFD) tools combined with model test data.

A better understanding of scaling effects between model and full scale is needed to reliably produce these large waterjet-powered ships. Since boundary layers do not directly scale between model and full scale, the scale effects of hull boundary layers on the waterjet performance need to be carefully considered. CFD tools and model tests will play major roles in developing these important scaling relations.

Aeration and/or emergence of the waterjet inlet may result in a sudden and possibly severe drop in shaft torque that can have a serious impact on propulsion machinery. Hull model tests under different sea states are needed to predict and minimize aeration and emergence occurrences. Potential aeration impacts on shafting and the waterjet components and structure need to be considered in the design. Methods and systems to minimize aeration and/or inlet emergence, such as ride controls and wave sensors, need to be developed. In addition, large high-power waterjets for ocean-going ships will operate for extended periods of time at near their maximum power ratings. This will necessitate careful attention to stresses and bearings to facilitate long life.

The large powers and high speeds involved could require large, heavy steering and reversing structures that can be on the order of the waterjet system weight itself using conventional jet deflection techniques. With a multiple waterjet arrangement, not all the units would require steering and reversing. Reversing may only be required at low speeds and powers, which would simplify the approach. This operational issue impacts the waterjet design and depends on the type of ship utilized. Steering can be effectively accomplished by means other than deflection of the jet, especially at high speeds. For example, high-speed ferries using "Interceptor" steering, a concept similar to an adjustable trim tab to impact hull flow, have demonstrated steering performance comparable to or better than that of ships equipped with steerable waterjets. In addition to reducing maintenance of waterjet steering gear machinery, alternatives such as

Interceptors eliminate the loss of thrust that results from using waterjet steering systems. Steering and reversing designs are critical elements of HSS waterjet design in light of the high speed and power application requirements.

Achieving the powering goals of up to 100 Megawatts will require waterjets to be developed that will absorb up to several times the power of today's most powerful waterjets. This will require an understanding of the mechanical design and fabrication aspects of potentially very large and high-power waterjet designs as well as understanding pump hydrodynamics. Casting limitations on physically large pumps may make alternate approaches, such as fabrication with separate blades and components, a more realistic approach.

5.3.4 Overview of Development Plan

The development of waterjet technology is evolutionary in nature. As a result, the plan is focused initially on advancing the axial waterjet technology from today's 10-13 MW size pumps to the 43 MW near-term pumps. This near-term technology is then the basis for subsequent evolution of the 100 MW far-term waterjets.

The tasks, time to complete each task, and costs associated with developing the needed waterjet technology are shown in Figure 5.3.4-1. Two stages of waterjet development, model testing, and analysis are shown to address both near-term and far-term waterjet technology. Costs shown are engineering estimates, based on the expected scope of testing and facilities required. This hullform peculiar program will require essential data from other technology development efforts such as powering (section 3.2), seakeeping (section 3.3), and maneuvering (section 3.4), as well as model test data for hullforms of interest (section 2).

Larger, higher power axial-inducer waterjets have not been pursued in recent times, and renewed development of this promising technology has high potential payoff. A near-term waterjet design for the 43 MW gas turbine is the next step in the state-of-the-art for axial-inducer waterjets. The technology needed to manufacture a pump optimized for the 40-50 knot design speed range will be produced. This represents an increase in axial-flow waterjet powering of about 3 times the present demonstrated capability, and will result in a unit with an impeller diameter in the likely range of 2.5 to 3.5 meters, slightly larger than the 1.8-2.0 meter impellers currently being manufactured for mixed-flow pumps. A 4-year development cycle is required to produce the full-scale near-term pump prototype.

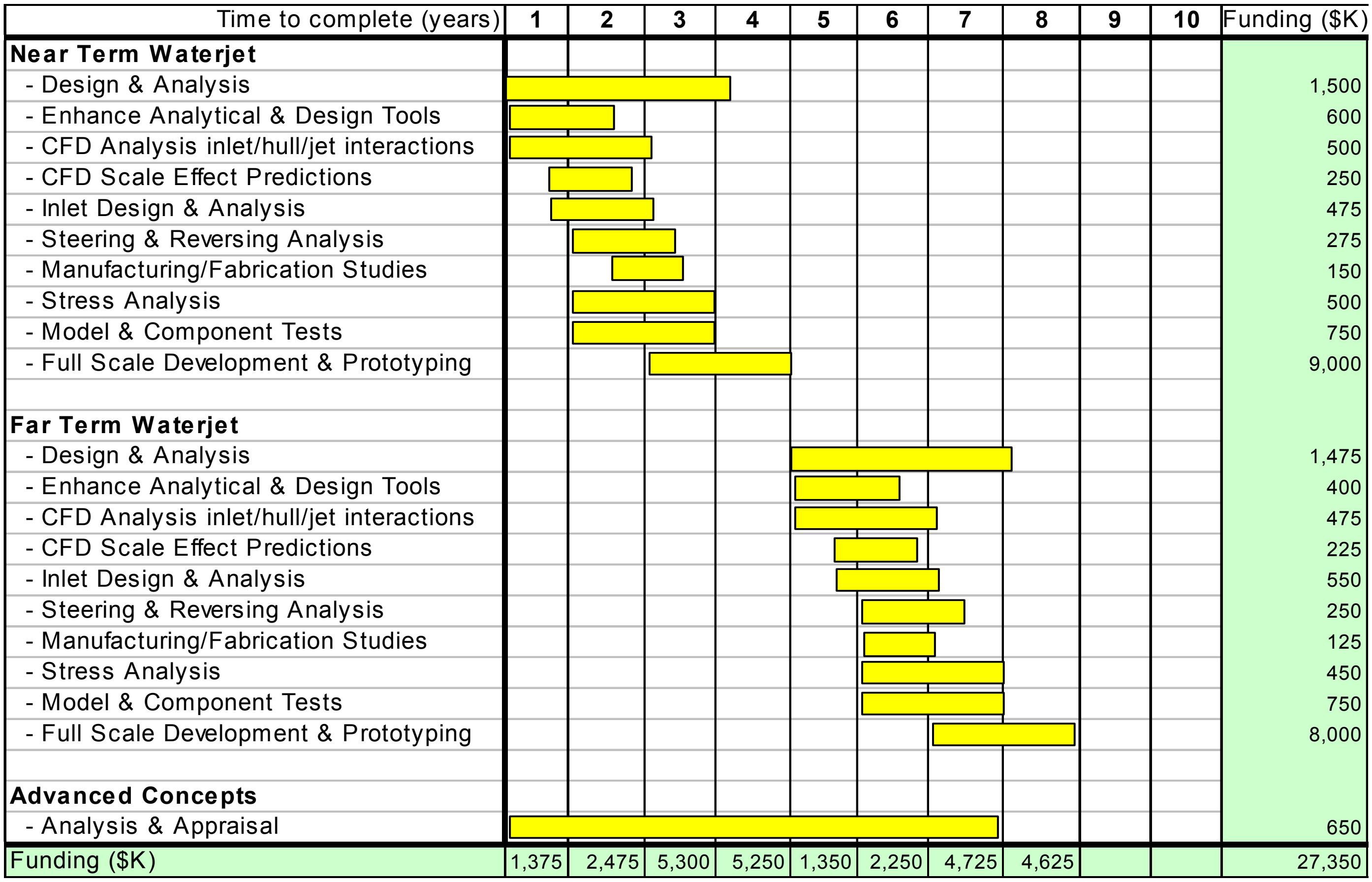


Figure 5.3.4-1: Waterjet Technology Development Plan

Steps in the development plan for the near-term 43 MW axial-inducer waterjet include:

1. Development would be aimed at a marinized gas turbine in the 40-50 MW power range that would be the likely power source. This power level favors ship designs having speeds in the 45-55 knot range and would represent a near-term technology for a very large waterjet.
2. Enhance analytical and design tools for the near-term speed and power range.
3. CFD analysis and prediction of inlet, hull, and jet interaction effects.
4. CFD development for full-scale design predictions and model to full-scale correlations.
5. Inlet analysis and design studies.
6. Steering and reversing gear analysis and design.
7. Mechanical design studies and manufacturing/fabrication analysis.
8. Scale-model testing to verify performance and cavitation scaling.
9. Full-scale design.
10. Stress analysis of all critical waterjet-related components.

11. Prototype testing on a ship of opportunity or installation on destination ship.

The far-term waterjet will be aimed at ships having speeds in the 50-70 knot range. To provide the requisite thrust for a ship of meaningful ocean-going size will require huge amounts of propulsive power. Waterjets capable of absorbing 90 MW of power will be required. Although the amount of power transferred will be double that of the near-term design, relatively small changes in waterjet size will result. As a consequence, the waterjet will become very power dense, with increasing design ship speed exacerbating mechanical and structural design challenges.

Steps in the development plan for the near-term 43 MW axial-inducer waterjet include:

1. Development would be aimed at a marinized gas turbine in the 90-100 MW power range that would be the likely power source. This power level favors ship designs with speeds in the 60-70 knot range and would represent a far-term technology.
2. Enhance analytical and design tools for the far-term speed and power range.
3. CFD analysis and prediction of inlet, hull, and jet interaction effects.
4. CFD development for full-scale predictions and model to full-scale correlations.
5. Inlet analysis and design studies.
6. Steering and reversing analysis and design.
7. Mechanical design studies.
8. Model testing to verify performance and cavitation scaling.
9. Full-scale design.
10. Stress analysis of all critical waterjet-related components.
11. Prototype testing on a ship of opportunity or installation on destination ship.

5.4 <u>Reduction Gears</u>

The weight of reduction gears represents a significant portion of the total ship weight due to the high installed power required for high-speed sealift ships. Technology development is required, as simply scaling-up existing designs to the desired power levels results in high weights.

5.4.1 Introduction

HSS designs use either a type of offset gear known as a locked-train double-reduction gear or epicyclic gears to transmit power from gas turbines to waterjets. The reduction gears are generally of the single-input, single-output type. All gears are non-reversing.

Offset gears are the most widely used type of gear for ship propulsion. These gears are well understood and available from numerous manufacturers. Offset gears are generally the lowest cost gearboxes due to the relatively simple machining and grinding required to produce the gear.

The epicyclic gearbox, on the other hand, is not widely used at sea. The epicyclic gearbox provides a very high power density (i.e., it is smaller than the offset gear) and typically weighs less as well. The primary disadvantage of the epicyclic gear is its increased cost relative to the offset gear. This is due primarily to the fact that the epicyclic gear contains more gear meshes than does the offset gear.

While conventional ships primarily use offset gears, when size and weight are important, as they are for the high-speed sealift ships, the cost disadvantages of the epicyclic gears become less of a factor. During the development of the near-term and far-term HSS designs, both types of reduction gears were used. As a result, requirements for two separate development paths presently exist, as no gearbox of either type exists that fully satisfies the reduction gear requirements established by the designs.

5.4.2 State-of-the-Art

HSS designs used both offset (parallel shaft) gears and epicyclic gears. Monohull and trimaran designs used offset gears, SES designs used epicyclic gears, and catamaran designs used both offsets and epicyclics.

Both offset and epicyclic gears have been built in power levels as large as the far-term needs of the present program. While the largest power units have not been for marine propulsion, it is important that the manufacturing facilities already exist for such large units. For marine propulsion, larger offset gears exist than do for epicyclic gears. Offset gears up to 50 MW per shaft have seen service, such as those for aircraft carriers. While these designs are significantly heavier than required for the high-speed sealift, Philadelphia Gear has designed a lightweight offset gear of similar power for FastShip Atlantic that weighs about two-thirds the weight of the older design. With a weight of about 1.0 kg/kW, this represents the state-of-the-art for marine offset gears.

Epicyclic gears offer the potential for significant size reductions and less weight. However, the additional complexity of these gears translates into higher procurement costs. As a result, epicyclic gears have not seen widespread use for ship propulsion. The largest marine units in-service to date have been on the order of 15 MW, although Cincinnati Gear offers a 25 MW epicyclic gear. These units are very lightweight at around 0.2 kg/kW. Cincinnati Gear designed a 30 MW epicyclic gear for the 3KSES program that had a weight of 0.13 kg/kW. However, these units all made significant use of aluminum for the housing. At power levels above about 30 MW, the use of aluminum may not be feasible. Any weight gains potentially due to the low density of aluminum are more than likely to be offset by the increased material necessary to provide the required stiffness. In the case of the 3KSES design, the design lifetime was undoubtedly shorter than that required by a ship designed to be commercially viable over years, if not decades, of service. Factoring in the issues associated with the use of aluminum for high power levels and potential design lifetime concerns could significantly raise the above weight levels. The 90 MW epicyclic gear built by Philadelphia Gear for use in a hydroelectric plant, with a weight of 0.88 kg/kW, is considered more appropriate to consider the state-of-the-art, although not designed for the harsher environment associated with ship propulsion.

5.4.3 Technology Goals

Near-term goals for both gearbox types are for lightweight designs capable of transmitting between 45 and 50 MW, while the far-term power requirement is 90 MW output.

Reduction gear weight is heavily influenced by the input power level, torque and reduction ratio. Figure 5.4.3-1 shows gearbox performance against power level for existing gears and HSS designs. Performance is represented as weight per 'torque', where 'torque' is simply the power divided by the output rpm. The HSS designs represent estimated design weights resulting from the final combination of gas turbine input power and speed and the design speed of the waterjet. Also shown on the curve are estimated trend lines for both offset and epicyclic gears.

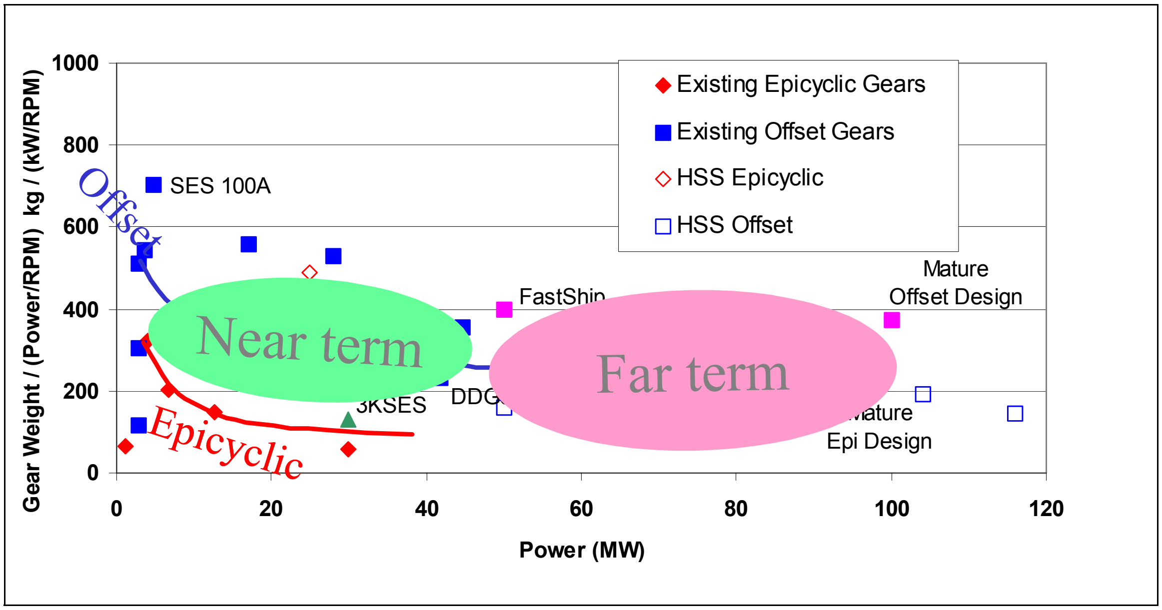


Figure 5.4.3-1: Reduction Gear Technology

In the near-term, the offset gears have weight per unit torque goals of between 200-400 kg/(kW/rpm), while the epicyclic gears have goals of 250-300 kg/(kW/rpm). Far-term goals are between 150-250 kg/(kW/rpm) for offset gears and 150-200 kg/(kW/rpm) for the epicyclic gears.

Figure 5.4.3-2 compares existing reduction gears with HSS designs on the basis of weight per unit power. In terms of weight per unit power, offset goals were 0.7-0.9 kg/kW and 0.6-0.7 kg/kW for the near and far-term goals, respectively. For epicyclic gears, the near and far-term goals were 0.6-0.7 kg/kW and 0.3-0.6 kg/kW.

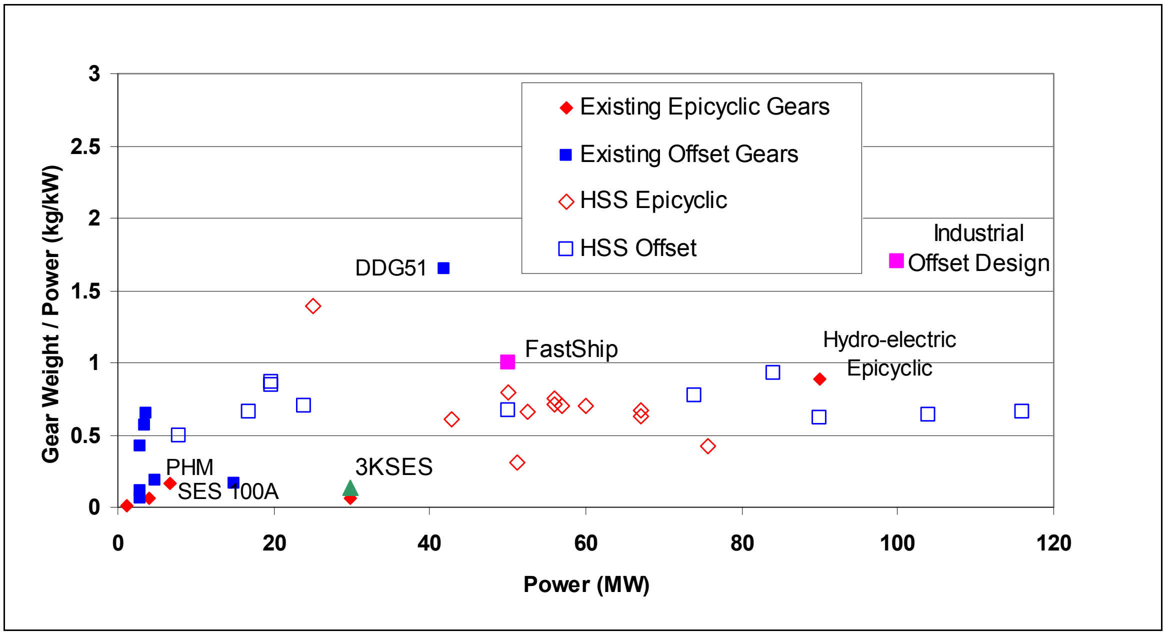


Figure 5.4.3-2: Weight per Power vs. Power

5.4.4 Overview of Development Plan

The principal focus for development for both offset and epicyclic marine gears is reducing the weight to the target levels. In terms of power output, similarly-sized gears have been built, if not for marine propulsion, at least for hydroelectric use. However, these existing designs are much heavier than what the fast sealift high-speed ship designs require. Therefore, significant effort, principally in materials development, is required if the goals are to be met.

Traditionally, commercial ship reduction gear designs have been built with cost as one of the most important considerations, if not the most important consideration. Size and weight usually have less priority. Military gears have particular requirements levied on them that make low cost less important. But in both cases, weight is not usually an overriding factor. However, for fast sealift to be both technically and economically feasible, the additional cost for lighter gears may be justified.

Major technology development in gear design optimization for minimum weight and materials improvements are required to achieve the target weight goals. Both offset and epicyclic gears will benefit from advances in these areas, although the payoffs may be different for each type. Design optimization is principally a matter of investing the effort to make reduced weight the critical factor. For example, the required lifetime operating duty cycle and environment needs to be rigorously examined to determine whether design margins can be refined. Materials improvement relates to the use of stronger and lighter materials as well as the manufacturing processes required to make use of these materials possible. For example, operating at higher temperatures reduces cooling and lubrication requirements, but this requires high-strength steels able to operate continuously at these elevated temperatures.

In the past, improvements in machining accuracy have enabled lighter weight gears to be built. With more precision machining, tooth contact area is increased, reducing resultant stress levels. This has been translated into smaller and lighter gear meshes, reducing the overall gear weight. However, the current ability of gear manufacturers to produce gears of very high dimensional accuracy and surface tolerance is such that it is unlikely that any significant improvements are likely without potentially very large investment. For example, Cincinnati Gear can already grind gears to an AGMA Class 15 quality on sizes up to 158 inches in diameter, where Class 16 represents a 'perfect' gear. Accordingly, the present plan does not account for any gains in this area.

Figure 5.4.4-1 presents a summary-level plan of technology required to achieve the specified near-term and far-term goals for both offset and epicyclic gears. Depending on the future direction of the HSS program, it may be necessary to develop only one type of reduction gear. There is a great deal of overlap in the development effort required; the primary differences only come into play when a specific design is being developed.

The plan shows that gears of both types can be available to meet the near-term goals in five years, followed by continued development to meet the far-term goals in ten years.

Time to complete (years)	1	2	3	4	5	6	7	8	9	10	Funding ($K)
Materials Development											
- Investigate High-Strength Steels											2,600
- Advanced Materials Treatments											4,000
- Fabricate and test large specimens											1,000
Offset Gear Development											
- Near-Term Design											400
- Near-term Prototype Fabrication and Test											2,500
- Far-Term Design											500
- Far-term Prototype Fabrication and Test											5,000
Epicyclic Gear Development											
- Near-term Design											500
- Near-term Prototype Fabrication and Test											3,000
- Far-term Design											500
- Far-term Prototype Fabrication and Test											6,000
Funding ($K)	500	2,500	3,500	3,000	4,000	1,000	4,500	5,000	2,000		26,000

Figure 5.4.4-1: Reduction Gear Technology Development Plan

Development of advanced high-strength steels and associated post-forging treatments to increase the overall strength and durability of the materials used for the gear mesh is required. If successfully applied to large gears, weight reductions approaching 25 percent may be realized.

Large gears are typically fabricated using 9310 steel. There are several advanced high-strength steels that are becoming available and are in use for smaller gears. These steels, such as Carpenter's Pyrowear 53 and Vasco's X2-M, provide higher strength and the ability to operate at higher temperatures. These steels provide the potential for designing large gears that are inherently stronger than existing gears. Further, the ability to operate at higher temperatures

reduces the requirement for cooling and lubrication, further reducing the overall size and weight of the gear mesh. This then contributes to reduced size and weight of the gear housing.

The technology to manufacture large HSS gears from these steels will be developed. In particular, means of producing large billets and fabrication methods for the large gears required will be developed. This will be a fairly lengthy process, as new processes will need to be conceived, developed, and implemented. Smaller specimens will be produced and tested to verify material properties. As the results of the scale-up become available, they can be integrated into the near-term designs of the offset and epicyclic reduction gears.

Advanced means of surface strengthening and finishing will be developed. Potential benefits include improved performance (higher strength), increased reliability and reduced costs. One such process is ausforming, developed by Penn State. Ausform finishing integrates heat treatment and hard finishing processes into a single phase. The process has been applied to much smaller gears than are necessary for high-speed sealift, but the results to date show significant improvements in both final strength and dimensional accuracy. While none of the individual processes involved with ausforming are in themselves new to the industry, the integration of these processes into a single operation will require development to be applicable to large-scale gears.

Existing gear manufacturing facilities will be developed, or at least modified, to provide the capability to apply advanced techniques such as ausforming to very large-size gears on the order of ten feet in diameter. Gear manufacturers already have facilities to produce these large gears, and even to perform some materials finishing at this scale. However, the combination of processes that ausforming requires is beyond their current capabilities. Facilities to achieve the desired capabilities will be developed. The scale of the required development is such that suitably treated large gears will not be available in time to support the near-term designs. However, the far-term reduction gear designs will be able to benefit from these advanced techniques.

The near-term designs should be able to take advantage of the availability of reasonably large billets of the advanced high-strength steels, but the far-term designs require larger gears of the same material that have gone through advanced finishing treatments such as ausforming. While the designs of the far-term gears can proceed on the basis of projected performance, an effort to fabricate and test large-scale specimens to verify performance is required. Material technology will be validated through production of very large billets or forgings of the advanced steels, ausforming the gear teeth to provide the desired strength, and then formal specimen testing of the final product to verify characteristics such as mechanical properties, dimensional accuracies and surface finish.

Both offset and epicyclic reduction gears are being carried forward in this development plan. It may turn out that only one type of gearbox is ultimately required, or that near-term missions will be satisfied by one type of gear and the far-term missions by another. As a result, at the present time, it is prudent to show development paths for both gears.

5.4.4.1 Near-Term Offset Gear Development

The near-term requirement for the offset reduction gears may be satisfied by commercial development in the next few years, at least in terms of the power absorption requirement. If this is the case, it is likely that the near-term plan described herein will be unnecessary. While the weight of the commercially-developed gear may not be as low as desired, it may prove more cost effective to apply the funds targeted for near-term development towards the far-term requirements, where the weight goals are more critical and more demanding.

Present offset gear designs for large marine use have not been optimized for minimum weight. Since gear weight does not represent a very significant part of the total displacement, the shipbuilder will tend to opt for a lower cost gear. This means that the gear manufacturer will not devote a great deal of time to reducing weight. Doing so requires engineering development and that adds to the cost to the buyer, who is unlikely to pay for such extravagance. Further, to minimize life-cycle costs, the gears are typically designed to operate without failure for the life of the ship. This increases the size and weight of the gear, since design margins will be large to eliminate problems associated with fatigue.

As the speed of the ship increases, weight becomes more and more critical in every element of the ship. This allows the costs associated with reducing weight to be more acceptable to the operator. In the present application, the operator is likely to be the military, whose primary interest is getting the cargo to the troops in the combat area. While it is desirable that the overall ship provides some degree of commercial viability, the primary emphasis on the military mission means that the costs associated with optimizing the weight of the reduction gear is a worthwhile investment.

Detailed weight reduction engineering analyses, including comprehensive finite element analyses of the entire gearbox design, will be performed to identify potential weight savings. The impact of the lightweight designs on manufacturing costs will be assessed. Design margins will be analyzed to determine where they may possibly be relaxed. It may be that reducing the design lifetime will provide weight savings, although achieving significant weight reductions via this path may require design lifetimes that are too short to be economically feasible. Further, the operating loads associated with very large, very fast ships operating in the open ocean will counteract to some extent the weight reductions available. Weight savings on the order of 20 percent may be possible through these approaches, with much of the weight reduction coming out of the housing (which may be as much as 50 percent of the total weight of an offset reduction gear).

The design of a prototype 43 MW gear suitable for the near-term mission will be developed. Representative design specifications (reduction ratio, operational and environmental loads, etc.) will be developed based on the near-term HSS designs. The design effort will incorporate the results of the weight reduction analyses as well as the use of advanced steels, providing the results of that investigation indicate that sufficiently large billets of suitable material would be available in time to support the fabrication schedule.

Using the FastShip Atlantic offset gear as a starting point, it is expected that the combination of the systematic weight reduction efforts and the possible use of the advanced steel may result in a weight reduction on the order of 25 percent for near-term offset gears, leading to a weight of perhaps 0.75 kg/kW.

A prototype 43 MW offset reduction gear, designed to meet the near-term mission, will be fabricated and tested under load by the manufacturer. The test setup will mimic the expected shafting and coupling arrangement between the turbine and the waterjet. The test will not accurately reproduce the full spectrum of loads associated with the at-sea application. Comparing strain gage and deflection measurements with those predicted for the test conditions will permit extrapolation to the expected at-sea loads. It may be feasible to consider an at-sea test of this gearbox, but this requires the availability of a ship powered by appropriately-sized turbines and waterjets. This is unlikely, unless a near-term mission ship is actually designed and built.

5.4.4.2 Far-Term Offset Gear Development

The far-term requirement is for a lightweight offset reduction gear suitable for operating with a 90 MW gas turbine. The weight targets are such that existing high-power offset gears are not suitable, and new designs must be developed.

A 90 MW offset gear that meets HSS weight goals will be designed. Further gains in weight optimization are expected, simply due to economies of scale. In addition to relying on weight optimization analyses, the use of the advanced high-strength steels as well as the improved materials processing (such as ausforming) will be required. Total weight reduction, relative to the FastShip Atlantic design point, will be on the order of one-third, leading to a weight of about 0.60 kg/kW.

A prototype 90 MW offset gear will be fabricated and tested by the manufacturer. As with the near-term prototype gear, it is unlikely that at-sea testing will be possible. The prototype gear will be heavily instrumented and the resultant stresses compared to predictions, allowing extrapolation to at-sea conditions.

5.4.4.3 Near-Term Epicyclic Gear Development

The development path for the epicyclic gear is essentially the same as for offset gears. Therefore, only important differences between the two paths will be identified.

It is unlikely that commercial development will produce a 43 MW epicyclic gear in the next few years. Therefore, unlike the offset gear, the cost associated with the near-term epicyclic gear will need to be funded by the Government.

Much of the weight savings for epicyclic gears achievable through weight reduction design optimization come from the gear housing. Epicyclic gear housing weight is a smaller fraction of the total gear weight than for offset gears, so the potential weight savings are proportionately less. Weight savings for epicyclic gears through optimization are estimated at 15 percent at best.

The design of the 43 MW epicyclic gear assumes the use of the higher strength steels. Overall, the weight savings achievable in the near-term are expected to be on the order of 20 percent vice perhaps 25 percent for offset gears. Epicyclic gears are generally somewhat lighter than offset gears of similar power, so a target weight of perhaps 0.75 kg/kW appears reasonable.

Prototype fabrication for the near-term epicyclic gear is essentially the same as for the offset gear. Costs are slightly higher simply due to the increased machining required to produce the sun and the planets compared to the offset gears.

The scope and effort of the prototype testing will be the same for the epicyclic gear as for the offset gear.

5.4.4.4 Far-Term Epicyclic Gear Development

The far-term requirement is for a lightweight epicyclic reduction gear suitable for operating with a 90 MW gas turbine. The weight targets are such that existing 90 MW epicyclic gears are not suitable, and new designs must be developed.

The design of the 90 MW epicyclic gears will incorporate the weight optimization efforts performed for the near-term design, including additional economies of scale. The use of high-strength steel as well as improved materials processing will be required. Total weight savings compared to current performance is 15 percent via weight optimization, and an additional 25 percent due to improved materials and processing. Several 90 MW epicyclic gears have been built for hydroelectric use, with a weight of 0.88 kg/kW. Applying the identified weight reduction goals, a final weight of perhaps between 0.55 and 0.6 kg/kW appears possible. This is at the high end of the desired level of performance, but it does not appear realistic to hope for further improvement.

Prototype fabrication of the far-term epicyclic gear is the about same as for the far-term offset gear prototype fabrication, again with the slight additional costs due to the additional complexity of the epicyclic gear design.

Prototype epicyclic gear testing has the same scope as for testing the far-term offset gear prototype.

The gear technology development plan will produce very lightweight marine reduction gears that satisfy the requirements identified for HSS designs. It is uncertain whether both or just one type will ultimately be required, especially with an eye towards the long-term, high-speed trans-ocean missions. Therefore, plans have been presented for both offset and epicyclic gears, as both have been identified as necessary to satisfy all of the HSS designs.

The plan will lead to a high level of confidence in the final products since full-scale gears will be fabricated and tested. While the test programs stop short of testing installed in a ship, in-shop testing will permit confident extrapolation to at-sea conditions. Not fabricating and testing the prototype gears could reduce the total cost of the development plan. This approach is worthy of

consideration principally for the near-term goals, as the weight targets assumed and the advances incorporated to achieve those goals are less demanding. However, full-scale prototype fabrication and testing is necessary to meet the more aggressive goals of the far-term gears. Analysis alone will not be sufficient to reduce risk to an acceptable level due to the combination of optimized weight designs, new materials, and new materials processing, combined with an operational scenario for which no previous design data exists.

Near-term commercial development for programs such as the proposed FastShip Atlantic project may produce offset reduction gears that come close to satisfying the near-term mission power transmission needs while falling short of weight reduction goals. Should this happen, the near-term offset gear development effort can be dropped. The additional weight gains achievable in the near-term would not justify the added expense. The related near-term materials work would still be required as it is an essential precursor of the far-term gear weight reduction effort. If this approach were implemented, the funding for the far-term design would need to be increased to incorporate the weight optimization task.

5.5 SES Lift Fans

5.5.1 Introduction

Large high-speed sealift SES will require lift fans with a combination of pressure and flow that represents the top end of the state-of-the-art in centrifugal fan design. However, several manufacturers produce fans that could meet all the requirements of the largest SES.

In addition to meeting the performance requirements, the SES lift fans must have as flat a pressure-flow characteristic as possible with a mild stall at low flow. These properties are necessary to allow the ship to benefit to the greatest extent from the SES concept.

5.5.2 State-of-the-Art

Lift fans suitable for the SES variants of the HSS missions are state-of-the-art. Fans meeting the pressure and flow requirements of the SES are commonly in use for industrial applications worldwide. These fans typically have welded steel impellers and housings. In order to minimize weight, the fan impellers could be fabricated from riveted or bolted aluminum, and the housing could be of welded aluminum. For marine applications, the shafts would be of stainless steel. Fabrication in aluminum is available upon request. Advanced composite materials are used in the most recent SES fan systems and offer considerable advantages.

Lift fans, required for the HSS study SES options, range from 2,000 KW in size to 8,200 KW in size. The largest study fan is characterized in Table 5.5.2-1. Also characterized is one example of a similar fan operating in the U.S. and of a fan quoted by the same manufacturer as available for order. The existing and quoted fans are of steel, but could be provided in aluminum with a weight savings of approximately 50 percent realized. It should be noted that the existing ABB fan characterized in Table 5.5.2-1 moves very high-temperature air. Thus, the flow rate and outlet pressures are high as compared to the SES fan parameters. Figures 5.5.2-1 and 5.5.2-2 compare state-of-the-art fan parameters with HSS design requirements.

Table 5.5.2-1: Comparison of Existing Fan Technology with High-Speed Sealift Requirement

Parameter	USN Vision & USA Vision Design	Pennsylvania Power & Light Power Station	Commercially Available to USN/USA Vision Design Specifications
Manufacturer	SES Design Synthesis Model (8 fans total)	ABB Fan Group	ABB Fan Group
Type	Narrow Contrifugal DWDI	Narrow Contrifugal DWDI	Narrow Contrifugal DWDI
Power – KW System Outlet Static Pressure – PA	8165 KW 3212	8940 KW 8867	9685 KW 3212
System Intlet Flow M^3/SR6	440M^3/sec	685M^3/sec (140°c)	440 M^3/sec
Fan Diameter	3.9M	3.75M	3.7M
Fan RPM	977	890	880
Fan Tip Speed M/sec	200	175	170
Fan Weight – Kg	16,800 (aluminum)	unknown	40,200 (steel) 10,000 (est./aluminum) 8,000 (est./composite)

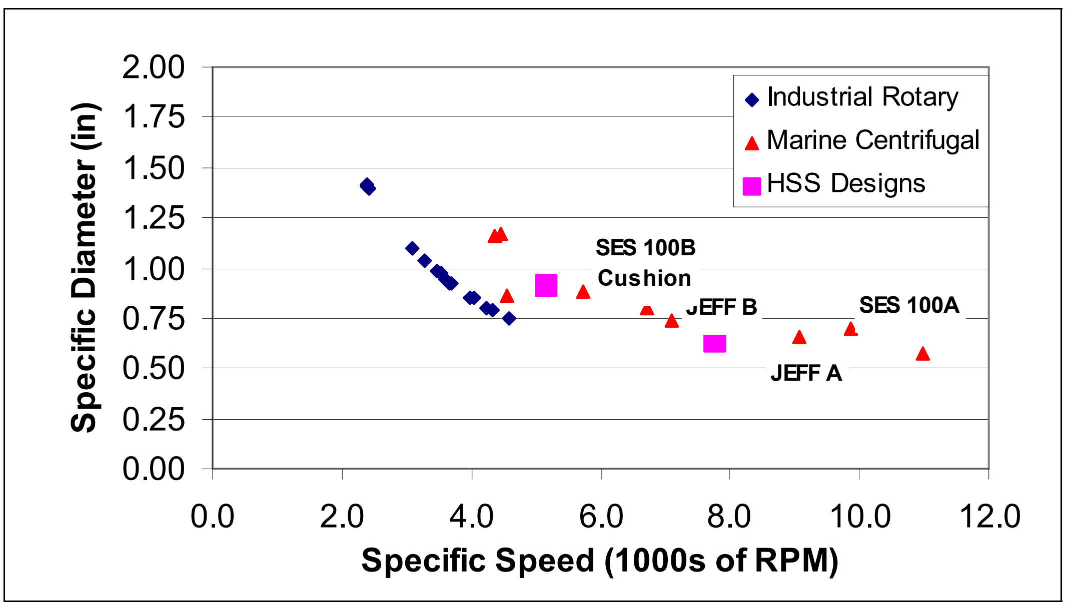


Figure 5.5.2-1. Comparison of Non-Dimensional Parameters of Existing and Design Fans

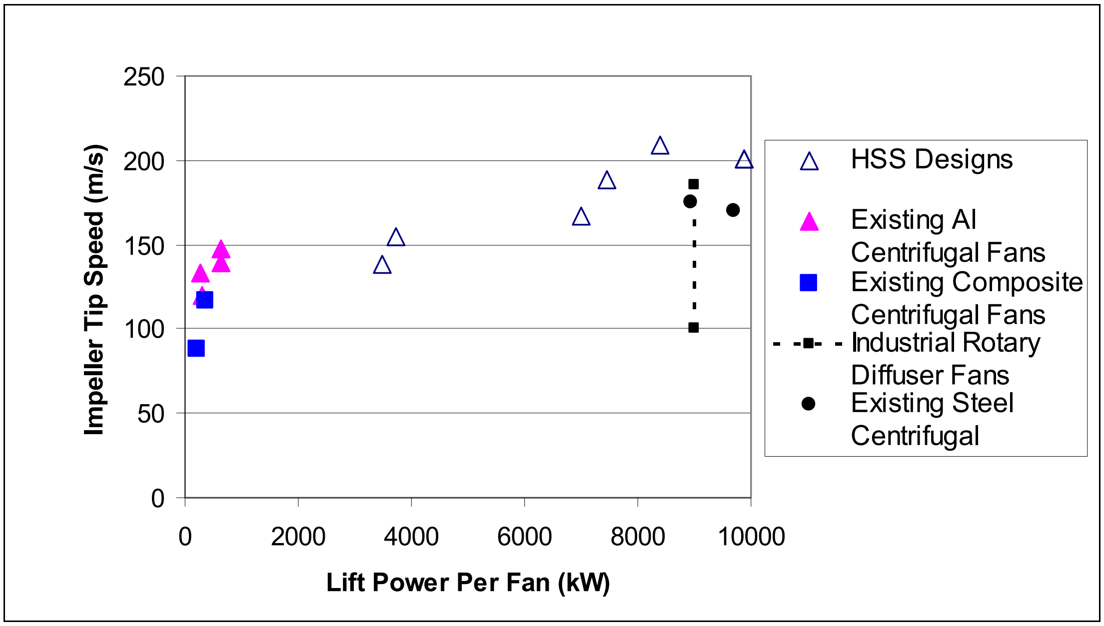


Figure 5.5.2-2. Comparison of "Power Density" of Existing and Design Fans

The requirements for all the SES lift fans for the large SES high-speed sealift ships studied in this project fall within the state-of-the-art for industrial centrifugal fans, and possibly for two-stage axial fans. Operation in a marine environment is not a major consideration for centrifugal fans. Some industrial fans are designed to operate in much more severe (corrosive) conditions. The use of composite materials for centrifugal lift fans has been amply demonstrated in a fleet of 9 SES MCM craft in Norway. Although the impeller diameter of these fans, designed by GLA in the U.S., was close to one meter, larger composite fans close to two meters diameter have also been built by UMOE in Norway. These BLA-designed fans have also been delivered to Finland where they have been installed in the Aker T2000 military ACV currently undergoing acceptance trials. UMOE states that they are able to design and build the same design in composite to four meters diameter. The composite SES fans have thousands of operating hours with perfect reliability and little maintenance. The UMOE fans incorporate a coating system which has completely protected the composite blades.

Of major concern to all fan manufacturers is the tip speed required to give the pressure required, which is about 2 m of water gauge, or 20 kPa., combined with the corresponding width of blade necessary to give the flow required. The combination of high tip speed and wide blade gives rise to high stresses that are difficult to meet in the structural design, especially when the stresses due to accelerations and gyroscopic loading from ship motions are added. This may be a limiting factor for steel and aluminum, or even titanium fans. However, the problem is much less severe for composite fans due to the lower density of composite materials.

Tip speeds of about 200 m/s appear to represent the state-of-the-art for centrifugal fan design. Fans to meet the requirements for the large HSS SES need tip speeds in the range of 168-200 m/s. Fortunately, one of the most promising industrial fan candidates appears to have the lowest

tip speed. Nonetheless, in this speed range, compressibility must be taken into account due to the high Mach numbers encountered.

Candidate manufacturers who have been contacted and have supplied information and data include: ABB, Aerophysics, Barron (a NYB company), and UMOE/BLA. Others who could not supply suitable fans include Chicago Blower, Howden Buffalo and Northern Blower. Aerophysics, at present, is able to design RD fans for SES that would then be made by selected manufacturers on a custom basis.

It can be seen that even the largest Innovation Cell Study SES designs have incorporated fans which are at the edge of, but within the limits of, state-of-the-art. Fan development will need to focus on reducing weight through the use of lightweight materials and on marinizing commercial designs as required.

5.5.3 Technology Goals

The primary technology goal will be to reduce the weight of existing large industrial steel fans by use of lighter weight materials. A weight reduction goal of 50 percent is required to meet the weight value utilized in the designs. Prior experience with centrifugal fans has shown no issues with changing material from steel to aluminum. The manufacturers of the BLA-designed composite fans for the latest Norwegian SES Minehunter and Finnish ACV state that a carbon fiber/epoxy 3.7-meter fan is within their capabilities to manufacture. The dynamic blade loadings of the 3.7-meter design fan are only 80 percent of those of the Norwegian SES fans.

The total lift fan weight for the largest far-term HSS designs is 134 MT. While it is understood from industry that this is achievable with aluminum and with composite materials assumed, use of existing steel fans would increase total ship weight by 187 MT. This represents a 1.4 percent increase in total lightship weight, or a 4.1 percent decrease in the 4,500 MT cargo load.

The near-term fan development goals are to refine the preliminary fan designs taking into account the marine environment. Weight and size of installation would be considered as well as the number of fans required. Drawings of the complete fan installation would be matched to the space available in the hull. In particular, the proposed UMOE/BLA composite fan design could be carried a step further.

Near-term goals are to:

- Seek the best combination of weight, space, power and performance.
- Pursue composite manufacture for very large fans.
- Refine the pressure and flow requirements.
- Study in more detail the fan performance characteristics during start-up and off-design operation.

Far-term goals include:

- Pursuit of advanced fan designs, including axial fans.
- Study new materials to seek the lightest weight fan designs, including advanced composites.
- Pursue and develop advanced fan designs to improve efficiency and reduce power, weight and space required, and to have pressure-flow characteristics suitable for SES operation.
- Demonstrate the sensitivity of the above improvements on the whole-ship design for selected SES.
- Conduct a model test program at small and large-scale for the leading candidate fans to demonstrate the performance predicted by the designers and manufacturers.
- Review manufacturing methodology for leading fans in the light of information developed by the above tasks.
- Study the possibility of using two axial fans in some applications of large SES, e.g., a combination of axial and centrifugal fans that might save space or weight.

In addition to total ship weight savings, lighter weight fans are desirable for reducing the gyroscopic loads on bearings and support structures and for reducing dynamic torque loads on the power train.

5.5.4 Overview of Development Plan

The application of surface effect ships for high-speed sealift requires development of high-performance lift fans with power outputs and tip speeds exceeding any previous SES application. Existing commercial centrifugal lift fans can satisfy near-term and far-term technology objectives, but some development is required to decrease weight and improve reliability in a marine environment.

The primary development need is to investigate and validate the use of high-strength lightweight material alternatives, principally aluminum and composites. Other important issues to be investigated include acoustics, blade erosion and corrosion, and stall-free and mild pressure-flow characteristics at design and off design operations.

The size requirement for near-term fans is well within the state-of-the-art, and minimal development beyond building and testing a prototype is required. The larger far-term fans are at the edge of the state-of-the-art and will require initial engineering studies and investigations prior to prototyping.

The development schedule and cost for HSS SES Lift Fan Technology is shown in Figure 5.5.4-1. Manufacturing costs provided by the fan suppliers are very similar for steel, aluminum, or composite.

Time to complete (years)	1	2	3	4	5	6	7	8	9	10	Funding ($K)
Near Term Lift Fan Development											
- Identify/Analyze Fan Alternatives											200
- Design Adaptation to Marine Environment											250
- Acoustic Studies and Tests											100
Fabricate/Test Near term Lift Fan											1,500
Far Term Lift Fan Development											
- Materials, Stress Analyses											750
- Engineering Design Development											1,500
- Fabricate/Test Far Term Fan prototype											2,000
Funding ($K)	200	350	1,750	1,000	2,000	1,000					6,300

Figure 5.5.4-1. SES Lift Fan Technology Development Plan

5.6 SES Seals

Figure 2.5.1-1 compares the size and speed of existing SES with near and far-term HSS designs. The largest commercial SES built is the 56-knot, 1,500-ton Japanese TechnoSuperLiner (TSL) HISHO (KIBO) launched in 1994. The largest military SES is the 45-knot Russian Dergach 700-ton missile patrol craft. As is the case for all high-speed craft, both near and far-term HSS SES designs are substantially larger than vessels produced to date. While SES and test craft have demonstrated higher speeds than those required for HSS missions, the combination of high speeds and much larger sizes of the HSS designs place heavy demands on seal technology. Significant development is required to produce the technology needed to manufacture seals for HSS SES concepts.

5.6.1 State-of-the-Art

The history of SES seal systems commenced early in the 20th century, long before the tests of the first British ACV test craft, the SR.N1, in 1959. The first cushion sealing system on the SR.N1 ACV utilized a "peripheral air jet" to maintain cushion pressure. The severe limitations of the peripheral air jet with respect to hover height and obstacle (wave) clearance became immediately apparent and renewed efforts to design a flexible seal (skirt) system were initiated.

Subsequently, the captured air bubble (CAB) ship studies initiated in the U.S. in 1960 also demonstrated the need for flexible bow and stern seal systems. Over the next ten years, dramatic advances in the development of flexible seal systems were made and demonstrated on operational ACVs and SES. The U.S. Government, primarily through U.S. Navy R&D programs, provided the major thrust for investigating the high-speed SES operational envelope and cushion sealing requirements.

The early seal systems developed by the British were basically designed to extend the peripheral air jet nozzle on a flexible curtain (Ref. SR.N1 ACV and Denny D1 SES test craft). It was at this point that seal system development started to reflect the operational requirements of the respective vehicles. The amphibious ACV required a fully-flexible, peripheral seal, whereas the

SES required essentially two-dimensional bow and stern seal systems. Stern seals for SES have typically received less attention than the bow seal and have generally exhibited minimal problems. Early seal configurations were complex and prone to early structural failure. However, as operational experience increased, the seal system configurations evolved and became less complicated. The natural inflation geometries of elastomer-coated-fabric materials were gradually accepted as the correct design approach with the seal shape contributing to the stability of the craft. The bag and finger seal designs were the final evolution of the flexible peripheral-jet nozzle. As the seal designs improved and the operational characteristics of ACVs and SES became established, the operating envelope was pushed to higher speeds and rougher surface conditions (i.e. rougher terrain and/or sea state). The service life, maintenance and cost effectiveness of the seals now became a dominant issue. The search for an elastomer-coated-fabric material that could withstand the high-speed marine (and for ACVs – terrain) environment became a major field of study.

Recognizing that there may be natural limitations to elastomer-coated-fabric technology, alternate seal system approaches were developed, model tested, and installed on operational test craft such as the XR-1D and the SES-100A1. Known as the “semi-rigid planing seals”, they represented a radical departure from the elastomer-coated-fabric bag and finger type seals that had been developed up to that time. The planing seal studies were conducted from 1967 through 1979 under U.S. Navy sponsorship and were aimed at lowering the risk for the 3,000-ton transoceanic SES (3KSES) program. The 3KSES was designed with full-length side-hulls to accommodate either a planing or bag and finger type seal. Due to the cancellation of the 3KSES program in 1979, neither the operational merits nor the deficiencies of these systems were fully established. The planing seal fitted to the SES-100A1 was not successfully developed due to a persistent series of mechanical failures. A large body of sub-scale test craft and full-scale seal component test data was amassed under the 3KSES program on the planing seal system.

The termination of the 3KSES program effectively ended further major research into the “high-speed” (60-100 knots) “high-cushion-pressure” (200-350 psf) transoceanic seal system development. However, the continuing Amphibious Assault Landing Craft (AALC) and follow-on Landing Craft Air Cushion (LCAC) programs in the U.S. provided a basis for continuing research of ACV bag and finger seals. Concurrently, the U.S. Navy redirected its SES development to a series of conceptual and model test studies of high length-to-beam ratio craft operating at significantly lower speeds (20-55 knots). The independent development of the Bell Halter SES (BH 110) for commercial and light military applications has been a useful platform for continuing seal system studies. In 1981, the U.S. Navy developed and tested the “Transverse Membrane Seal” (TMS) on SES 200, a stretched BH 110. The concept employed a basic elastomer-coated-fabric membrane, stiffened transversely by fiberglass battens. The TMS seal showed promising results in model tests, but inconclusive results from full-scale tests on the SES 200 as the program was terminated following damage to the seal before testing was completed.

European activity in the seal development area for both ACVs and SES has focused on improving the operational life of the bag and finger seals. Bag and finger seals were used exclusively by the British Hovercraft Corporation (BHC), one of the world’s foremost manufacturers of ACVs, on their entire range of vehicles. This system has over 300,000 operational hours of experience since the basic version was introduced in the early sixties. The

basic bag and finger design is also employed on the U.S. Navy's LCAC and the U.S. Army's LACV-30 amphibious lighter. Other versions of the design have been successfully employed on the SES-100B 100-ton high-speed test craft and on the British Vosper/Hovermarine HM-series of SES. There were over 100 HM-series craft built and successfully operated over a wide range of commercial ferry applications worldwide. Full-depth finger seals (i.e. the finger component minus the air supply bag) have been employed successfully on the BH110 series and, more recently, on Scandinavian SES, including the Norwegian MCM and Patrol Craft, the Norwegian Skjold SES patrol craft (which is capable of speeds in excess of 60 kts), and the Swedish SMYGE Patrol Craft.

Various SES designs have called for bow and/or stern seal retraction as an operational mode. Seal retraction is desirable for off-cushion, slow-speed operations to minimize overall craft drag and protect the seal from excessive hydrodynamic loading. The SES-100A in its original configuration was also designed for partial-cushion operations (i.e. side-hulls partially immersed) using retraction mechanisms for both bow and stern seals. Operational testing of this system aboard the SES-100A was not successful and the concept was abandoned.

Seal technology to support development of large transoceanic SES is based on more than forty years of operational experience with both military and commercial SES up to 1,500 tons, military and commercial ACVs up to 300 tons, and a large body of tests data and analysis on seal systems for high-speed, high-cushion-pressure, ocean-going SES (e.g. 3KSES, SES-700 and related studies). An analysis of the history of seal development shows the following three basic design approaches have been used to meet cushion seal requirements for ACVs and SES:

- Flexible Membrane Seals
  - Bag and finger (SRN-series, SES-100B, LCAC, LACV-30)
  - Loop segment (Vosper VT1 & 2, HM-series)
  - Full depth finger (SES 200, Norcat)
  - Loop-pericell (AALC, Jeff A)

- Semi-Flexible Reinforced Membrane Seals
  - Stay-stiffened membrane (SES-100A)
  - Stay-stiffened membrane (XR-1C)
  - Transversely-stiffened membrane (SES 200)

- Semi-Rigid Planing Seals
  - Bag and planing surface (XR-1D)
  - Bag and segmented planing surface (SES-100A1)

The development phases of the SES concept from the early sixties through mid-seventies was aimed towards large, high-speed, ocean-going SES. Towards that end, the U.S. Navy invested a considerable effort in the development of seal systems capable of sustaining the high-speed, high-cushion-pressure environment. Two systems were evaluated quite thoroughly under both sub-scale and full-scale test conditions. These systems were the bag and finger bow seals and the semi-rigid planing bow and stern seal system.

The operational dynamics of the bag and finger seals were well understood by the mid-seventies as a result of extensive scale-model tests and full-scale operational data analysis. However, the relationship between high over-water speed (60 to 85+ knots) and the wear rates of elastomer-coated-fabric fingers was not clear. Several component tests and full-scale environmental test programs were conducted during this period to obtain wear data. Full-scale environmental testing was conducted on candidate fingers for the 3KSES to determine wear rates. The data were correlated with operational experience (albeit on the lower end of the speed and cushion pressure scale) from the SR.N4 cross-channel ferry service. The dominant effect of over-water speed on elastomer-coated-fabric wear rates is readily apparent. While the effects of internal (cushion) pressure on finger wear was not fully characterized in these tests, the data indicated a lower rate of wear increase as a function of increased internal pressure as compared with speed induced wear. Further tests and analysis conducted under U.S. Navy sponsorship characterized the mechanisms of finger flagellation and failure modes. These tests also measured fingertip accelerations on the order of $3{,}000^{+}$ g and up to 500^{+} hz. The overall evaluation of the bag and finger seal for the high-speed 3KSES application was satisfactory with respect to dynamic stability and performance, but unacceptable due to the high wear rates of the finger elements. It should be noted, however, that finger life at the lower speed and cushion pressure regimes, as clearly demonstrated by the heavy-duty commercial operations across the English Channel, were well within the acceptable range.

The semi-rigid planing seals were developed primarily as an answer to the high-speed seal-life problems experienced by the bag and finger seals. The smooth, high-strength, glass-reinforced planers appeared highly resistant to flagellation effects and high-speed water damage. The planing-seal tow-tank test data for the 3KSES indicated promising performance in the areas of drag reduction and seal contributions to stability. However, operational experience aboard the XR-1D and, later, the SES-100A1 test craft highlighted significant structural problems involving attachments and vulnerability to self-induced damage (e.g. planers interacting and damaging each other). The methods employed to “fix” or “band aid” the test craft seals increased their weight considerably and, hence, aggravated their operational problems. The overall evaluation of the semi-rigid planing seals tested through 1979 was that they suffered from inherent structural design problems that could not be easily overcome.

Through 1979, the emphasis on “high speed” for transoceanic SES such as 3KSES dominated seal technology development in the U.S. Subsequently, the emphasis has changed with the redirection of the U.S. Navy’s SES technology studies towards the high L/B, slower craft (i.e. 35-55 knots). Currently, both high and low L/B SES are operating with simplified finger seal systems at the lower speed regimes with satisfactory results. The finger wear experience has been good, with over 1,000 hours of operation at speeds up to 55 knots and cushion pressures to 100 psf. The finger elements have always been considered frangible (i.e. remove and replace) items. Replacing the lower portion of the finger or cuff has proven to be an economic way of extending finger life. Typically, fingers can be removed and replaced without drydocking these craft.

The failure mechanism of the coated-fabric materials under the ACV/SES environment was well understood by the mid-seventies. However, the experience of wear on the LCAC and LACV-30

skirt systems is based upon materials developed over twenty-two years ago. Understandably, finger wear rates on these craft tend to be high, primarily due to their abrasive, amphibious operations over concrete ramps, rough terrain, and ship well-deck transitions. Note, however, that a relatively simple geometry change made to the LCAC skirt with the introduction of the LCAC Deep Skirt has more than doubled the finger life even though there has been no change to the material used to construct the skirt fingers. Vast experience on skirt wear can be found in reference to the British Hovercraft Corporation's SR.N4 70-kt passenger and car cross-channel ferry, which operated between 1968 and 2001, and on the U.S. Navy LCAC program, which now has over 70,000 hours of accumulated operating hours. Significant advances have also been made over the past twenty years in the development of elastomer compounds for applications outside the SES and ACV areas. New, very lightweight material from a U.S. supplier has recently been tested at the labs at GKN Westland U.K. and underway on LCAC at ACU 5. Results to date are very encouraging. Further study of these technology improvements and collection of current operational data would be an important contribution to the seals design database.

5.6.2 Technology Goals

Bag and finger seals have evolved as the most reliable system within the context of current SES and ACV operations. However, it is recognized that operational parameters cannot be directly scaled from current experience to HSS concepts without further development. With respect to the HSS seals requirement, the major question that needs resolution is the ability of the seal elastomer-coated-fabric in immediate contact with the water surface to provide adequate operational life. At present, there is no adequate means of extrapolating seal wear (life) characteristics from existing data. There is clear evidence of elastomer-coated-fabric finger deterioration with increased over-water speed. In addition to the moderate to high-speed regime, the HSS mission requires operation at significantly higher cushion pressures. A better understanding of the effect of increased cushion pressure on finger wear rates is needed. An alternative seal configuration, such as the TMS seal, may reduce seal/surface flagellation and, hence, improve seal life. However, additional development of this system is required to resolve structural design issues before it can be seriously considered as a candidate seal system for the HSS mission. The major thrust of the HSS seals development plan must, therefore, be directed at resolving the finger wear rates and developing structural seal configurations to obtain acceptable operational life requirements.

The goal of the seal technology development effort is to produce the technology needed to manufacture bow, stern, and transverse seals to meet HSS SES performance objectives and provide acceptable reliability and maintainability qualities.

5.6.3 Overview of Development Plan

HSS seal technology will be developed using a combination of analysis and testing. As the seals are essentially two-dimensional, the basic cross-section can be evaluated using simple force balance analysis based upon the defined cushion and seal pressures. The two-dimensional geometric analysis will take into account both the pressure balance and weight of the seal components to obtain equilibrium at full deployment. Mathematical models of the primary seal

components and their respective load paths are useful tools to analyze maximum stress and load concentrations. Having arrived at a balanced geometry, the resistance of the system to hydrodynamic drag forces will be investigated by analytically deforming the seal (i.e. immersion) at maximum speed. The operating envelope will be evaluated for various combinations of seal components (e.g. the finger height and width, seal pressure). A key design consideration for the stern seal is rapid response to waves to minimize cushion leakage at the higher speeds. This requires careful design of the bag air supply and exhaust system to ensure that when the seal is compressed, the bag pressure is dissipated and then rapidly replenished to restore the seal to the nominal deployed position. Alternative designs for the stern seal will be examined.

Small-scale, static models of the seals will be used to verify the geometric proportions of the design and highlight problems. The scale-model test rig will include an air supply that is a scaled representation of the design bag and cushion pressure (not necessarily the flow rate).

A retraction system is needed for both bow and stern seals to minimize drag and potential damage to the seal during off-cushion operations. Retraction also allows for the easy replacement of sacrificial cuffs at the lower portions of the seals that see the greatest wear. Typically, bow seal retraction is more difficult because the seal (e.g. bag and finger type) is not conducive to full retraction. However, partial retraction of the seal is relatively straightforward as far as the multi-lobed bag is concerned. The full-finger seal, however, presents obvious problems with regard to retraction and requires further development. An operational representation of a retraction system designed for the 3KSES was employed on the modified SES-100A test craft. While this planing seal had significant structural problems, the retraction system stood up quite well to operations.

Stern seal retraction has been successfully accomplished on several SES for both off-cushion operation and to increase side-hull immersion for ship pitch control and added waterjet inlet submergence to minimize air ingestion at certain operating conditions. Design consideration to account for “snatch” loads due to stern seal motions against the retraction straps in rough sea conditions is essential.

The applicability of a transverse seal will be determined by speed-power and motions requirements. If a transverse seal is required, the development process and retraction system would be the same as for the bow seal.

The best candidate for the HSS bow seal is a full-depth finger design having an upper and lower segment. The rationale for this design approach is based upon use of simple identical modular components that are readily maintainable or replaceable to extend operational life. Furthermore, there is no air supply required for this type of bow seal.

An alternate bow seal design for the HSS mission that deserves consideration is the TMS type. The TMS seal requires further development to solve the problems of transverse modularity and lateral wave contouring. However, the superior performance characteristics predicted for this type of seal are worth pursuing further.

The best known stern seal candidate is a multi-lobed bag with wear strips at the seal/water interface. The stern seal is fed via a boost fan that takes cushion air and increases the pressure approximately 20 percent. The primary function of the stern seal is to minimize the cushion air leakage under all operating conditions. Consequently, the seal must be responsive to wave-forms passing through the cushion. These types of stern seals can be prone to flutter, wherein the velocity of the air passing under the trailing edge can cause an unsteady state at the surface, causing the seal to oscillate. Typically, this is more of a problem at model-scale and in steady-state (calm water) conditions. Flutter can be corrected by adding devices to the wear strip to break-up the air flow patterns. Alternative designs for the stern seal will be examined.

The best transverse seal candidate is a bag and cone arrangement. The seal could be fed via a boost fan to increase the cushion pressure the required amount. An alternative seal type for the transverse seal is a full-finger seal.

The plan for developing seal technology for the HSS bow, stern and transverse (if applicable) seal systems is presented below. The tasks, time to complete each task, and costs associated with developing the needed seal technology are summarized in Figure 5.6.3-1. Seal technology development will focus on structural design, material requirements, maintainability, and produceability.

Time to complete (years)	1	2	3	4	5	6	7	8	9	10	Funding ($K)
Design & Development											500
Performance Verfication											1,100
Structural Design											500
Material Selection											1,100
Reliability & Maintainability											200
Produceability											200
Operational Verfication											3,400
Funding ($K)	1,400	2,700	2,900								7,000

Figure 5.6.3-1: SES Seal Technology Development Plan

The 3KSES database will be utilized as a foundation for the structural design approach. The structural design approach will consist of:

- a static loads analysis to define major load paths and stress concentrations of candidate seal designs subjected to a range of internal (seal/cushion) pressures and external hydrodynamic loads.
- a dynamic loads analysis using predictive techniques developed and correlated with test data under previous SES/ACV programs to analyze the effects of rapid seal redeployment (snap loads) and seal/cushion pressure transients for HSS seals.
- a bow and stern seal structural design using existing 3KSES data as the basis for the seal structural design for the HSS with appropriate selection and scaling of components and attachments to meet the seal system requirements. The structural design shall include the retraction mechanisms and all attachment fittings and hardware.

The material requirements analysis for the HSS seals will include elastomer-coated-fabrics, attachment fittings and hardware, and retraction mechanism materials.

The analysis of the elastomer-coated-fabric materials is the most important area of study. Elastomer-coated-fabric materials technology developed under the 3KSES program will be extended to take advantage of technology developments over the past ten years.

The material requirements for the attachment fittings and retraction mechanisms are basically state-of-the-art and can be based upon a large database of previous testing and analysis. Slight extension of 3KSES technology is required to reflect operational experience with seal attachment methods and hardware developed under the AALC and LCAC programs and retraction mechanisms developed for the SES-100A and SES-100B to select materials and systems for the HSS concepts.

Candidate elastomer-coated-fabrics for HSS SES applications will be tested to fully characterize a seal material including:

- mechanical properties tests utilizing standard testing procedures (Federal Standard Methods) to verify the mechanical properties of candidate seal materials.
- small-scale environmental tests to test candidate material samples under simulated HSS SES environmental conditions. Small-scale environmental tests are primarily used for comparison testing (i.e. known control seal material vs. candidate) to select the best candidates for further evaluation.
- large-scale environmental tests to simulate the full-scale operational environment of the HSS SES seals, including internal pressure (cushion/seal) and external hydrodynamic loading.

Seal system structural design test requirements will be based upon a careful review of existing full-scale component test data and operational experience with both high-speed test craft and long-term commercial operations. HSS bow and stern seal configurations will be tested aboard a sub-scale vehicle to verify attachment configurations and functional operation of the system.

The maintainability of the HSS seals is a major design consideration, which will be addressed from the outset of this plan both in the design configuration and the structural design of the system. The seal system will be analyzed with respect to installation and removal of major components and dock-side repair capabilities. It is essential that maintainability be built into the seals design. The tasks conducted under this area will include studies of maintenance techniques and repair philosophy, and evaluation of the candidate seal systems. A Mean Time Between Failure (MTBF) analysis will be conducted for bow, stern and transverse (if applicable) seal systems. The analysis will be further broken down into the primary components of the seal (e.g. bag system, finger system, etc.) to determine failure modes and the overall life expectancy of the system. MTBF data will be developed from both operational experience (test craft, commercial experience) and environmental test data from the seal material test and evaluation activity. The maintenance cost of the seal systems will be a specific focus of attention under this plan. The need to "dry-dock" the ship for seal maintenance will be considered vs. dockside maintenance

and the ability to maintain the seals internally (i.e. access through the wet-deck and plenum areas) while in the stowed or retracted position. The analysis will include a time, labor and services study on specific seal maintenance procedures with projected cost profiles.

Seal produceability is also a vital element in development of HSS seal technology. The capability of producing large runs of the selected HSS seal materials while maintaining high standards of quality control will be assessed in an elastomer-coated-fabric produceability analysis. The processes and techniques of seam bonding will be evaluated with respect to reliability and cost. Seal attachment fittings produceability, centered on the primary attachments of the seal to the hull structure and the internal seal module-to-module attachments, will be analyzed. The candidate designs will be reviewed with respect to minimizing dissimilar components and for simplicity of manufacture. Finally, retraction mechanisms produceability will be analyzed to address the ease of manufacture and assembly of retraction mechanism designs and the integration of retraction systems with ship systems (i.e. power supply, mounting, control, etc.). As part of the produceability studies, the projected manufacturing cost estimates will be identified for the major seal subsystems with supporting rationale.

Additional details of the seal technology development plan follow.

5.6.3.1 Seal System Design and Development Tasks

1. Conduct state-of-the-art trade-off studies of bow and stern seal configurations for the HSS.
2. Analyze bow and stern seal air supply requirements and define ducting and venting requirements.
3. Design and fabricate complete bow and stern seal models of selected seal system candidates.
4. Prepare design report detailing candidate bow, stern and transverse (if applicable) seal features and rationale for geometric proportions. Define baseline parameters and seal/hull/retraction mechanism interface requirements.
5. Conduct state-of-the-art survey of candidate retraction systems for the bow, stern and transverse (if applicable) seal systems. Conduct design studies and prepare trade-off analysis of candidate retraction systems.
6. Prepare design study report detailing candidate bow, stern and transverse (if applicable) seal systems.

5.6.3.2 Seal System Performance Verification Tasks

1. Prepare preliminary performance predictions and seal behavior profiles within the HSS operational envelope for the candidate bow, stern and transverse (if applicable) seals suitable to characterize seal performance in the areas of resistance, motions and stability.
2. Define test and instrumentation requirements for seals hydrodynamic testing and evaluation. Prepare test plan for tow-tank test series to characterize seal performance in the

areas of resistance, motions and stability. Test plan shall also include operations under partial seal damage conditions to analyze reduced performance characteristics of the seals.

3. Conduct bow, stern and transverse (if applicable) seal tow-tank tests in conjunction with side-hull hydrodynamic tests.
4. Correlate model test data with analytical predictions of seal system performance. Prepare full-scale HSS seal performance predictions.

5.6.3.3 Seal Structure Design Tasks

1. Conduct a static loads analysis of bow, stern and transverse (if applicable) seal systems to define the primary load paths within the seal structure and examine the effects of high transient internal and external loading.
2. Conduct a dynamics loads analysis of bow, stern and transverse (if applicable) seal systems under simulated seal behavioral patterns to predict the transient loads such as those due to spike pressure in the seal bags and hydrodynamic slamming loads.
3. Prepare bow, stern and transverse (if applicable) seal structural designs with sufficient detail of retraction mechanisms, attachments and fasteners for sub-scale and/or full-scale component prototype fabrication.

5.6.3.4 Seal Material Selection Tasks

1. Analyze seal materials requirements based upon the defined HSS operating environment and the loading predictions for the primary seal components.
2. Conduct state-of-the-art survey of seal system materials. Compare SOA with HSS materials requirement and define technical deficiency areas.
3. Prepare specification for candidate HSS elastomer-coated-fabric materials. Survey industry for prototype development capabilities.
4. Fabricate prototype candidate elastomer-coated-fabric materials suitable for standard testing and seal component environmental testing.
5. Define test plan for seal material test program including basic material characterization and SES environmental testing. Environmental testing will be conducted at both small (sample)-scale and large (component)-scale.
6. Conduct elastomer-coated-fabric test program.
7. Analyze test results and prepare summary of full-scale operational characteristics. Prepare seal life and MTBF predictions utilizing test results and operational experience data from available sources.
8. Prepare seal material selection analysis report.

5.6.3.5 Reliability and Maintainability Tasks

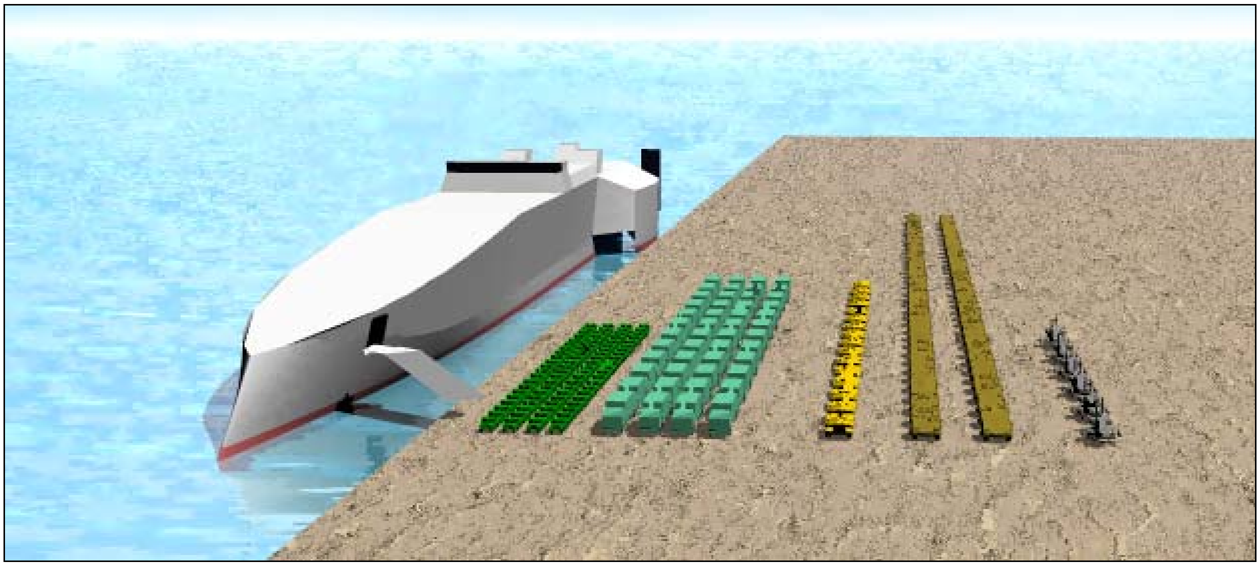
1. Conduct an FMEA and an operations safety analysis for the HSS bow, stern and transverse (if applicable) seal systems candidates. The analysis shall be closely coordinated with the tow-tank model test program. The analysis will consider various forms of bow, stern and transverse (if applicable) seal failure and combinations thereof with respect to ship safety and operability.
2. Prepare a maintenance analysis report which defines maintenance techniques and procedures for the HSS bow, stern and transverse (if applicable) seal systems. The analysis will include estimates of time, service equipment, and personnel requirements to accomplish periodic servicing and major repair activities.
3. Prepare a MTBF analysis for major seal system components based upon empirical data, where available, and qualitative projections.
4. Conduct a cost analysis study of primary maintenance activities utilizing data prepared under the above tasks.

5.6.3.6 Produceability Tasks

1. Conduct a produceability analysis for the bow, stern and transverse (if applicable) seal systems.
2. Prepare a cost analysis for the major seal system elements. Project costs for prototype and production quantities based upon industry evaluation of preliminary structural designs.

5.6.3.7 Seal System Operational Verification Tasks

1. Design and fabricate full-scale, land-based, test rig capable of simulating HSS bow, stern and transverse (if applicable) seal attachments and air supply system.
2. Test full-scale HSS seals to measure seal loads.
3. Develop sub-scale bow and stern seal system models for intermediate-size SES utilizing the HSS design criteria. The designs should be configured to scale HSS proportions to the maximum extent practical without compromising the safety or performance of the intermediate-size SES.
4. Conduct verification tests on intermediate-size HSS seal systems to confirm performance and durability under operational conditions.



6.0 CARGO HANDLING SYSTEMS

The technology and the ship-integration of cargo handling systems have two separate and significant impacts on the HSS mission. The weight and size of the cargo handling subsystems will trade-off against ship speed and/or payload. The onload and offload efficiency of the systems will affect the mission cycle time for the port-to-port delivery of cargo. Both must be evaluated and an overall impact on mission capability determined when choosing the cargo handling system.

6.1 Cargo Securing

6.1.1 State-of-the-Art

The sea transport of military vehicles is complicated by the wide variety of vehicle types and mixes used by the military. The variance in external dimensions and construction of the many vehicle types has led to adoption of the multi-point tensional tiedown system currently in use. This system incorporates steel cable or chain assemblies with hook/shackle terminals and mechanical tensioning and adjustment devices. The tiedowns are currently secured to the cargo decks utilizing standard steel deck fittings, which are welded to the stiffened plate steel decks. The tiedowns are heavy, cumbersome, and are time-consuming to install, adjust and remove.

Commercial pressures have led the high-speed ferry industry to simpler, lighter approaches. For example, INCAT high-speed ferries use a lightweight, fabric strap tiedown system. These straps secure the vehicle's wheels to tiedown points on stiffened plate aluminum decks. Furthermore, all vehicles are not secured on these high-speed ships that operate in coastal/sheltered waters. The stable nature of the ship, along with a high coefficient of friction deck surface, eliminates the need for securing vehicles in all but the most extreme operating conditions. All vehicles located on sloped decks are secured. These tiedowns are not capable of holding heavy military vehicles on long voyages.

6.1.2 Technology Goals

The primary goal is to develop lightweight securing systems that are compatible with heavy military vehicles and the advanced deck structure concepts discussed in section 4.0 while minimizing the mission cycle time for HSS ships. Weight reduction is needed to avoid adversely impacting the speed/power performance of HSS ships. Of equal importance is the time required to load and unload the vessel since long loading and unloading times at the terminals will negate the transit time advantage of a high-speed ship.

6.1.3 Overview of Development Plan

Securing systems will be developed in conjunction with the lightweight deck structure (section 4.0). It may be fully or partially integrated into the structure or decks. Fittings shall be designed, manufactured, installed/integrated in lightweight deck prototypes and tested for expected load criteria and proof-test requirements.

The tasks, time to complete each task, and costs associated with developing the needed lightweight securing systems technology are shown in Figure 6.1.3-1. Costs shown are engineering estimates, based on the expected scope of testing and facilities required. This program will be integrated with lightweight structural technology by structural concepts (section 4.4).

Time to complete (years)	1	2	3	4	5	6	7	8	9	10	Funding ($K)
Cargo Securing											5,000
- prototype design											300
- hardware testing											200
Ship Arrangements for Cargo											
- prototype design											1,500
- hardware testing											3,500
Ramp Systems											
- prototype design											1,300
- hardware testing											500
Funding ($K)	2,800	2,800	1,700								7,300

Figure 6.1.3-1: Cargo Handling Systems Technology Development Plan

6.2 <u>Ship Arrangements for Cargo Handling</u>

6.2.1 State-of-the-Art

The internal arrangements of decks and ramps, and the number and location of onload/offload ports, have a significant impact on the time to load and offload vehicle cargo. The arrangements are also a trade-off against the weight penalties of multiple ports, ramps, and re-configurable decks.

The capability of modern CAD systems to provide rapid 3-D representation of internal ship configurations and vehicle configurations is commonplace. Software simulation models of vehicle movements have been used by NAVSEA to evaluate the trafficability of new MSC sealift ships. Similar modeling of MHE vehicles is being used in T-AKE design studies.

NAVAIR has developed a "man-in-the-loop" software simulation of weapons cargo handling aboard CVN(X) which incorporates actual human movements/functions along with the movements and functions of onboard cargo handling equipment.

Surveys and characterizations of port and pier facilities in remote locations have been done for USTRANSCOM and MARAD.

6.2.2 Technology Goals

The goal for this technology is to develop a comprehensive, integrated capability to model and simulate men and vehicle movements onboard HSS ships during onload and offload and to populate the simulation model with data accurately reflecting the timeline and tasks necessary to accomplish onload and offload.

6.2.3 Overview of Development Plan

Existing models that evaluate pieces of the onload/offload process will be integrated and calibrated with existing ship load/unload operations to simulate movement of men and vehicles onboard the ship. The load/unload time for HSS ship hullforms and configurations will be evaluated.

The tasks, time to complete each task, and costs associated with developing the needed technology are shown in Figure 6.1.3-1. Costs shown are engineering estimates, based on the expected scope of testing and facilities required.

6.3 Ramp Systems

6.3.1 State-of-the-Art

MSC and commercial RO/RO ships are equipped with various combinations of sideport, stern, and quartering onload/offload ramps. Sideport ramps typically comprise a deck-extension platform deployed outboard with a ramp extending from that platform to the pier, parallel with the ship. Generally, sideport ramps on military ships can be deployed and removed by shipboard cranes. Stern ramps typically extend aft, but may be slewed to port or starboard to land on an adjacent pier. Stern ramps are permanently attached (hinged) to the hull and are deployed by their own winch/cable system. A quartering ramp is fixed at an angle, usually at the stern, allowing it to land on a surface to the side of a ship.

All MSC and commercial sealift ramps are fabricated of steel. These ramps are designed to high factors of safety to meet Mil Standard and commercial regulatory organization rules and can weigh upwards of 200 tons. The ramps are designed to be landed on a level, horizontal, static

surface. The newest MSC ship ramps have some tolerance for motions and misalignments with the landing surface. However, the ramps are still very limited in their ability to sustain torsional loads resulting from uneven loading across the ramps' landing pads. There is on-going Navy R&D for floating Roll-On/Roll-Off Discharge Facilities (RRDF) to better understand ramp dynamic loads and develop modifications to reduce these loads.

Most commercial high-speed ferries have very short ramps designed to interface with purpose-built docking facilities or rely on land-based ramps for load/offload. These very lightweight ramps are usually built of aluminum.

Other lightweight ramp-like technology has evolved outside the marine field for applications such as portable bridges to allow military vehicles to cross rivers.

6.3.2 Technology Goals

Ramps designed to rapidly load and offload cargo are critical to accomplishment of HSS missions. The majority of the HSS study missions have a requirement to offload "in-stream", which describes an ability to land the ship's ramp on a beach, an RRDF located close to shore, or a floating or elevated pier extending from the beach. This requirement affects the configuration of the ramp as well as its strength and loading. Ramp configuration is also hullform dependent. Current technology ramps are heavy, resulting in adverse impacts to machinery and fuel weights. Development of lightweight ships' onload/offload ramp technology that is compatible with slender HSS hulls and advanced structural concepts is a technology goal.

6.3.3 Overview of Development Plan

Advanced lightweight ramp configurations will be developed for representative HSS designs addressing hullform-unique ship-to-pier, ship-to-beach, or ship-to-RRDF issues. The application of lightweight materials and structural concepts to onload/offload ramps will be integrated with the lightweight secondary structure developments discussed in section 4.0.

Ramp design concepts for reducing dynamic loads on ramp structures may be critical to HSS ships, where the "brute force" approach of multi-hundred ton ramps in current use may not be viable. Applicable results from on-going Navy RRDF R&D to reduce ramp dynamic loads will be integrated into the advanced ramp concepts developed.

A representative lightweight ramp will be fabricated and tested. This program will be integrated with the lightweight structural technology included in Structural Concepts (section 4.4).

The tasks, time to complete each task, and costs associated with developing the needed ramp technology are shown in Figure 6.1.3-1. Costs shown are engineering estimates, based on the expected scope of testing and facilities required. This program will be integrated with the lightweight structural technology included in Structural Concepts (section 4.4).

7.0 OTHER RELATED TECHNOLOGIES

7.1 Drag Reduction

Significant hullform technology development for displacement hulls and Surface Effect Ships (SES) is required to meet HSS mission speed and range objectives. Displacement hull powering improvements are obtained through use of very slender hulls to significantly reduce wavemaking resistance. While wavemaking resistance is reduced, the increased wetted area of the slender hulls results in increased frictional drag. Unlike lower speed ships, frictional drag of these slender hulls often greatly exceeds wavemaking drag. By contrast, SES derive their powering advantages through use of an air cushion to reduce hull wetted area, and hence frictional drag.

Many technologies have been proposed to reduce the frictional resistance due to the flow of water over a hull's surface. Investigations of these technologies have been conducted by many organizations over the past 30-40 years. While the physics of drag reduction have been demonstrated, none of the approaches have been found to be suitable for marine applications. Two of these technologies, which involve injection of drag reducing substances or micro air bubbles in boundary layers, have demonstrated significant frictional drag reducing potential in idealized laboratory experiments. Analytic models of the phenomena, backed by test data using laboratory-scale models (~6m in length) of simple shapes (e.g. flat plates), have been developed. Very significant frictional drag reduction has been demonstrated in these idealized conditions. Major technical issues remain such as the effectiveness of these technologies on complex ship-like shapes at ship scales in a marine environment, the amount of substance needed, and design of reliable, workable injection systems. While further basic research is needed to fully understand these technologies, continued investment from outside the sealift arena is anticipated to determine their potential and practicality outside the laboratory.

If these technologies are scaleable to ship-like proportions, their potential for high-speed sealift displacement ships is great, particularly for the higher speeds and longer ranges of inter-theater concepts. Intra-theater concepts will also benefit from substantial reductions of installed power, machinery plant weight, and machinery plant cost, but the higher speeds and longer ranges of the inter-theater concepts will result in disproportionately large reductions in fuel weight. While these technologies also apply to SES, the benefit is much less due to the lower wetted area of these vessels while on cushion.

The magnitude of these reductions for one of the far-term inter-theater displacement concepts is evident from Table 7.1-1. The characteristics shown for the drag reduction case are very simplistic in that they are only the direct effects and do not include the effects of design iteration needed to produce a fully-balanced design. However, substantial reductions in displacement, machinery plant size and weight, and fuel weight are evident.

Table 7.1-1: Impact of Drag Reduction Technology

	Without Drag Reduction	**With Drag Reduction**
Displacement (mt)	27,000	~24,000
Speed (knots)	55	55
Range (miles)	8,700	8,700
Payload (mt)	4,500	4,500
Machinery plant	**4x90 MW**	**3x90 MW**
Machinery weight (mt)	**1,900**	**1,500**
Fuel weight (mt)	**11,000**	**8,750**

The HSS Technology Workshop experts concluded that while frictional drag-reducing methods had great potential, development of suitable systems would require more than ten years to produce viable ship systems. Consequently, these technologies fall outside the far-term technology threshold adopted for the HSS designs produced. However, the great potential for these technologies for high-speed sealift ships warrants their continued support. In particular, at-sea demonstration of these technologies on a ship of 90-100 m length at speeds of 20 knots or more is recommended as an intermediate goal for transitioning drag reduction technology to sealift ship applications.

7.2 Composite Shafts

Power transmission shafts for ships are typically manufactured from steel. Steel shafts are heavy and require thru-life maintenance to counter galvanic and corrosion effects. While lightweight, non-metallic composite drive shafts are widely used in the aerospace industry, aerospace power/torque requirements are much lower than in the marine field. However, there is a limited marine market for composite shafting, particularly for high-speed ferries and fast patrol craft where weight is important. Composite shafts are in service in over 100 ships and craft at power levels below 10 MW. Military applications include the Swedish SES test craft SMYGE (2.6 MW), Norwegian OSKOY class MCMVs (1.4 MW), and Norwegian Skjold (6 MW). Composite shafts for HSS concepts must be capable of absorbing much higher power and torque.

The U.S. Navy has developed a mature technology to design and support fabrication of composite shafts for the range of powers and torques needed for HSS ships. Technology development includes design, fabrication, and full-scale land-based tests of a 10 m long 37 MW/2,500 kNm shaft section including related couplings. By comparison, commercial state-of-the-art composite shafts are capable of absorbing about one-sixteenth the torque of this 37 MW shaft section. While full-scale at-sea demonstration of this shaft on a large Navy auxiliary (AOE) was planned in the early 1990s, the plan was not implemented. Development of high-power, high-torque composite shaft technology has continued to address critical issues such as shaft/coupling/bearing sleeve joints and shaft strength. These advances have included fabrication and full-scale land-based testing of additional high-torque shaft systems, Figure 7.2-1, to validate shaft to coupling joint strength, shaft to bearing sleeve joint strength, load sharing predictions, and predicted failure modes. However, in-service experience with high-power, high-torque composite shafts is minimal.

Figure 7.2-1: State-of-the-Art High-Power, High-Torque Composite Shaft

In addition to reduced weight, composite shafts offer a number of potential benefits to ships including reduced corrosion and galvanic effects, reduced magnetic signature, reduced bearing loads, increased vibration dampening, and lighter weight. While none of these attributes is critical to development of HSS ships, the significant weight reductions possible with composite shafts make this technology attractive for HSS applications.

Shaft weight is a relatively small percentage of total ship weight. Consequently, composite shaft technology is not considered essential to the viability of weight-sensitive HSS ships. The relatively mature state of composite shaft technology, combined with the significant weight reduction possible, make composite shaft technology attractive to reduce emptyship weight of HSS ships and improve transport efficiency. However, operational validation of this technology is needed to reduce risk to acceptable levels. HSS power transmission requirements are 43 MW/1,500 kNm per shaft for near-term designs and 90 MW/3,500 kNm per shaft for far-term concepts. At-sea validation of a high-power, high-torque composite shaft is recommended since such a demonstration would be representative of near-term shafts while simultaneously serving as a suitable large-scale technology validation model for far-term concepts.

8.0 SUMMARY

The High-Speed Sealift Innovation Cell has produced ship designs for projected missions of interest to the U.S. Government to assess technology development requirements for the near-term (+5 years) and far-term (+10 years). Intra-theater missions with speeds of 40-50 knots, payloads of 450-1,500 tonnes, and ranges of 800-1,500 miles were included as well as inter-theater missions with speeds of 40-70 knots, payloads of 4,500-12,000 tonnes, and ranges of 4,000-10,000 miles. Designs were produced to a consistent set of standards for monohull, catamaran, trimaran, and SES hulls. Technology projections for these designs were taken from the High-Speed Sealift Technology Workshop held at the Naval Surface Warfare Center, Carderock Division in October 1997. In addition to advanced hulls, other technologies essential to these missions include advanced structures and materials, lightweight fuel-efficient gas turbines, reduction gears, waterjets, SES seals, and SES lift fans. The resulting designs varied from small intra-theater ships displacing a few thousand tonnes to inter-theater ships with displacements in excess of 50,000 tonnes.

The capabilities needed from each of the technologies to produce these designs were compared with the technical state-of-the-art for those technologies to define the necessary near-term and far-term technology enhancements. Estimates of the time to develop and rough order of magnitude development costs were made for each of the technologies. The goal of this technology development effort is to bring the individual technologies to a level of maturity sufficient to lower risk to levels appropriate to ship design and construction. Technology development plans for each of the technologies were provided in earlier sections of this report. Figure 8-1 is a summary of those plans.

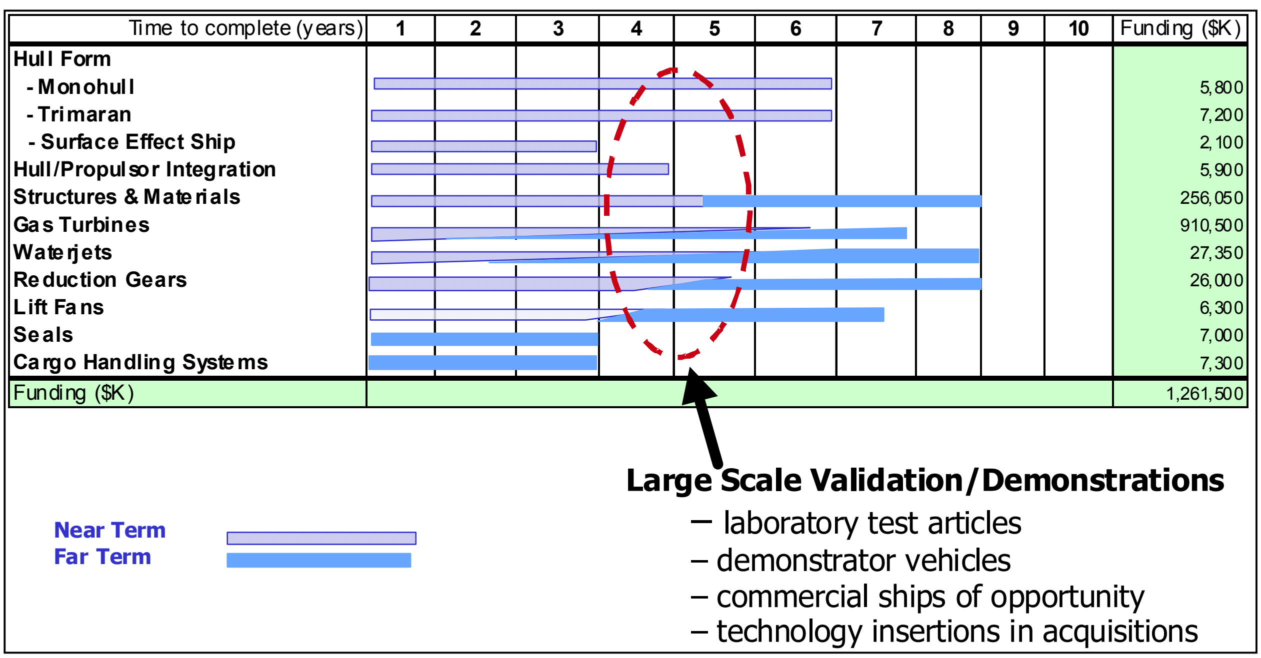


Figure 8-1: Comprehensive Technology Development Plan

These development plans are comprehensive, with no allowance for market-driven technology development that may occur through commercial initiatives. Some technology development in critical areas is expected to meet anticipated commercial needs. For example, development of large gas turbine technology is highly likely for aerospace, industrial, and commercial marine projects. While such commercial technology development efforts will potentially reduce the need for Government investment, elimination of this investment is not expected since there is some risk that the commercial efforts will either not come to fruition or the commercially-derived capabilities will fall short of the capabilities needed to meet the more demanding military requirements (e.g. duty cycle, sea state operability and structural loads, ambient air and water temperatures, maintenance philosophy). Consequently, the potential existence of these commercial efforts is identified, while the cost reductions that might result have not been shown.

The comprehensive plan shown in Figure 8-1 contains some necessary redundancies since the specific need for some of the technologies depends on other technology choices. The choice of hullform technology has a particularly large impact on requirements for other technologies. For example, development of far-term SES hulls requires development of SES-specific lift fan and seal technologies. Alternately, monohull and trimaran hulls require development of different reduction gear technology than SES or catamarans. Hullform choice may be strongly influenced by mission parameters other than speed, range, and payload. Other characteristics, such as length, beam, and draft, vary considerably among the four hullforms considered. While most of the designs produced were within the required limits, significant military advantage may result from the differences in proportions of the hullforms. For example, the shallow draft possible with an SES on-cushion may prove compelling to expand port access, particularly for intra-theater missions. Since such decisions cannot be made with certainty prior to commitment to specific long-term objectives, the redundancies have been identified and retained at the individual technology level. However, it is unlikely that the full matrix of technologies will be developed. Choices between alternatives will likely be made to further focus the technology development effort and reduce cost. A hullform-specific plan for displacement hulls (monohull and trimaran) is shown in Figure 8-2, while the SES plan is shown in Figure 8-3.

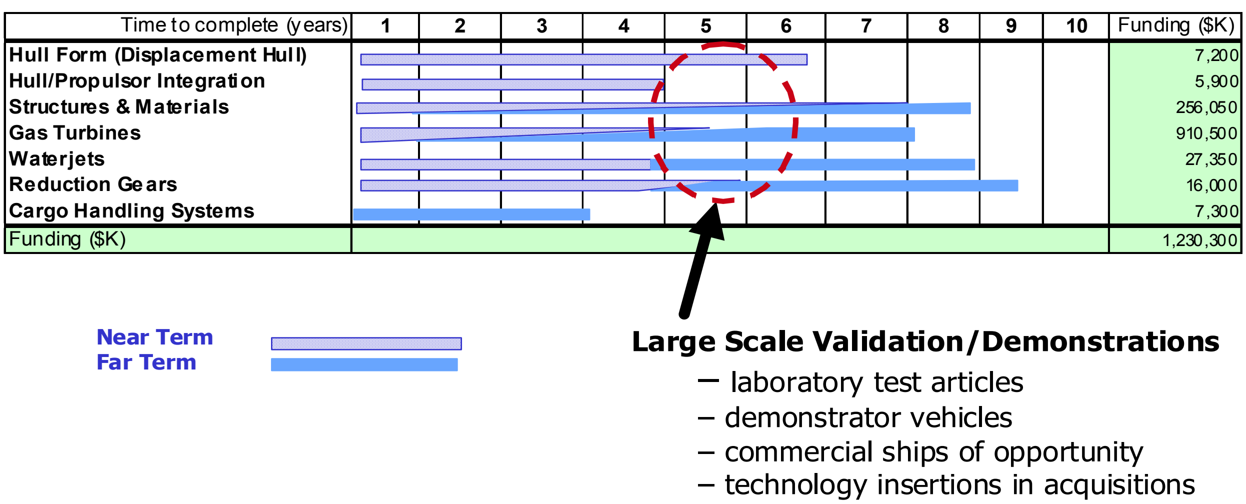


Figure 8-2: Comprehensive Technology Development Plan for Displacement Hulls

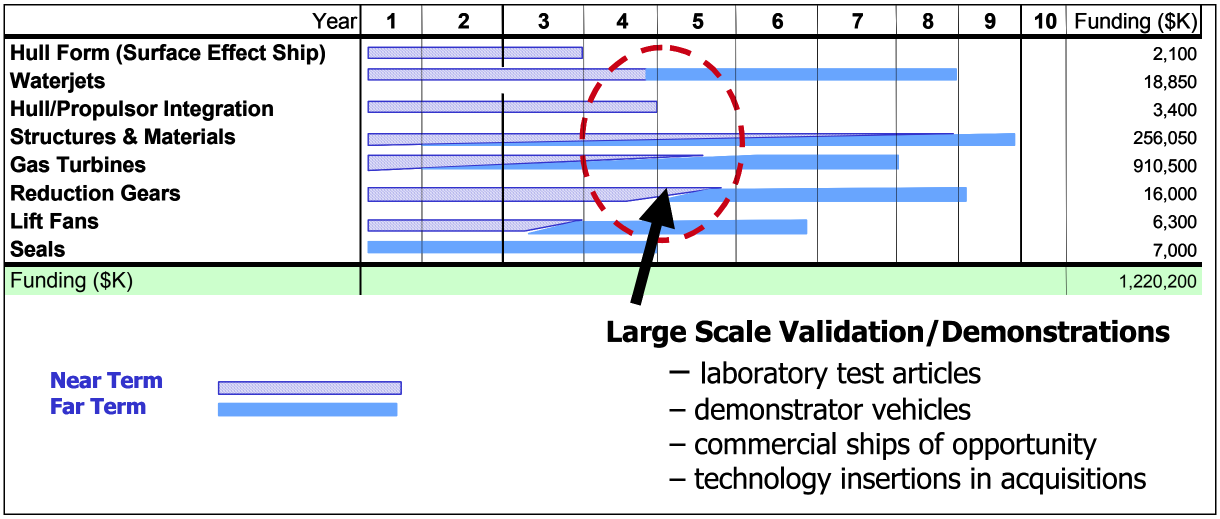


Figure 8-3: Comprehensive Technology Development Plan for SES Hulls

Several of the plans for development of individual technologies involve significant increases in scale from current technology levels. For example, near-term waterjets will require absorption of over twice the power of today's largest waterjets, while far-term power requirements are four times current levels. Similarly, far-term trimaran hulls displace over twenty times as much as the largest existing trimaran ship. Comparable increases in scale exist for advanced structures, gas turbines, reduction gears, and SES seals. Validation testing of large-scale specimens of these advanced technologies is included in the individual plans to validate the technologies, enhance technical credibility, and reduce technical risk to levels suitable for ship construction. This validation testing can be accomplished for most of the technologies through land-based testing or at-sea testing in suitable existing or specially-constructed ships. The costs associated with these large-scale tests are high. Costs associated with fabrication and testing of large-scale test articles accounts for 83 percent of the structures and materials cost, 28 percent of the gas turbine costs, and 46 percent of the gear cost. Insertion of selected technologies such as lightweight structural components (interior decks, ramps, composite deckhouses), composite shafts, or reduction gears into design and build projects may reduce the R&D costs of these technologies, albeit at some increase in acquisition cost and programmatic risk.

The advanced hullform technologies needed to achieve perceived mission requirements can only be validated through construction and operation of large prototypes of the advanced hulls. Costs for building and testing advanced hullform demonstrators such as the RV Triton trimaran demonstrator, Figure 8-4, have not been included in this plan. The Triton project was focused on de-risking an advanced hullform for a 30-knot, 4,000-tonne combatant mission. Validation of the hullform and structural technology was judged to be required at not less than 60 percent of the full-scale size and at speeds over 20 knots to reduce risk to acceptable levels. Construction cost of the 1,100-tonne RV Triton was about $20,000,000, while the two-year trials effort cost an additional $10,000,000. While not strictly applicable to HSS technology development requirements, the Triton example is indicative of the level of effort required to validate hullform technologies for advanced concepts such as the slender HSS displacement hulls or high L/B ratio SES.

Figure 8-4: RV Triton, Trimaran Hull Technology Demonstrator

The goal of the process defined by the 1997 High-Speed Sealift Technology Workshop was to determine the technology development requirements to support projected high-speed sealift requirements. Prospective mission requirements and ship concept designs for those missions have been developed by the HSS Innovation Cell project. This Technology Development Plan defines the level of technology required, as well as the cost and time to develop, for the technologies essential to realization of HSS ship concepts.

9.0 ACKNOWLEDGEMENTS

The continued support, direction and guidance provided by the High-Speed Sealift Executive Steering Committee and its Action Officer Group during the course of this study is gratefully acknowledged.

The contributions of Mr. Christopher Broadbent (UK MoD) and his team of UK MoD Graduate Engineers to this study is especially recognized. Their participation through the Navy International Programs Office Exchange Scientist and Engineer program contributed greatly to the project.

Band, Lavis and Associates is acknowledged for substantial contributions to this study through characterization of machinery and SES hullform technologies as well as technology integration for all hullforms.